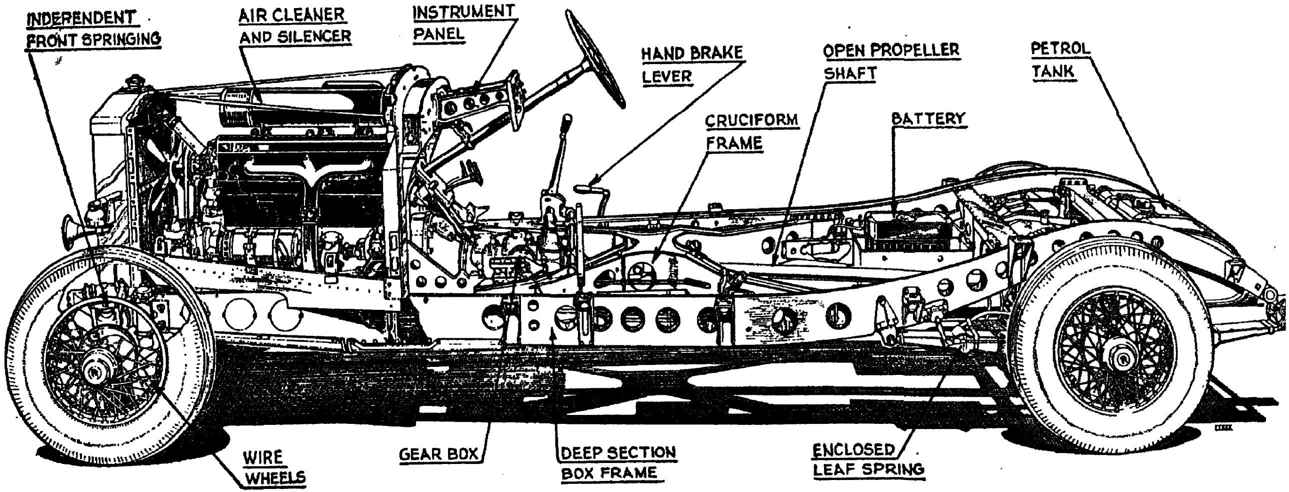

Frontispiece.

The Rolls Royce Chassis.

MOTOR MANUALS
A SERIES FOR ALL MOTOR OWNERS AND USERS

VOLUME III

THE MECHANISM OF THE CAR

ITS PRINCIPLES, DESIGN, CONSTRUCTION AND OPERATION

BY

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Fourth and Revised Edition
Reprinted

LONDON
CHAPMAN & HALL, LT
37, ESSEX STREET, W.C.2.
1946

Reprinted from standing type.

The paper and binding of this book conform to the authorized economy standards.

Printed in Great Britain by C. Tinling & Co., Ltd.
Liverpool, London, and Prescot

Bound by G. & J. Kitcat Ltd., London
Flexiback Binding.

PREFACE TO FOURTH EDITION

THIS series of small motor manuals was introduced originally for the use of the car owner, driver and mechanic desirous of obtaining fuller information than was possible from the popular motor manuals covering the whole subject of motor cars. Subsequent experience has indicated that these volumes have also become more extensively adopted in schools and technical institutions catering for the motor engineering student and apprentice so that the later editions have been extended a good deal beyond their original scope to meet the requirements of the latter classes of reader, whilst at the same time endeavouring to adhere to their original elementary plan.

The present edition includes much new material relating to the progress that has been made during the past two or three years notably in connection with the subjects of motor vehicle transmission systems and including accounts of semi-automatic systems, overdrives, underdrives, the "Electric Hand," turbo-flywheel, "Hydra-matic," Cotal and other modern innovations. The sections on independent front and rear wheel drives and suspensions, clutches, brakes and steering systems have also been revised and extended. Additional information has also been given on modern tyre developments and some further analytical information included in the section on "Traction and Braking Principles."

As a result of the revision over eighty pages of new matter and seventy new illustrations have been added.

Unfortunately, owing to the loss of all of the revision illustrations through enemy action, it has not been possible to carry out, in full, the author's original plan of revision although some of the loss has been made good in this volume.

A. W. JUDGE

Farnham, Surrey.
1942.

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CHAPTER I

THE CHASSIS IN GENERAL

Introductory.—In the present volume it is proposed to consider, in a general manner, the mechanical components of modern cars of different types. The subject of the engine has already been dealt with in the first volume of this series so that these considerations will include the frames, transmissions, brakes, suspension systems, steering gears, controls and similar items.

Before proceeding further, it will be useful to indicate the different classes of motor vehicle in common use, and also to refer to promising alternative types and those in the experimental stage. As we shall see, later, modern motor-car practice has settled down, at the time of writing, to almost rigid lines, namely with the radiato and engine at the fore end, and with a purely mechanica transmission to the rear driving wheels. The presen volume will therefore naturally give much greate prominence to this type of road vehicle.

Types of Automobile.—As we have mentioned, th majority of modern motor-cars, and a large proportion o commercial vehicles, employ the purely mechanical metho of transmitting the engine power to the road-wheels ; shall refer to this popular system, in future remarks a the Mechanical Type. There are, however, other kinds o road-vehicle, and each of these has its particular applica- tion and advantages. For the present we shall confine ou reference to a brief mention, only. Enumerating th different available kinds of road vehicle we have :

(1) Petrol-engine vehicle with mechanical transmission o power to the rear driving wheels ; and steering on th front wheels. A clutch, gear-box and differential ar invariably employed.

(2) Petrol engine vehicle with mechanical transmissio of power to the front driving wheels, the latter bein

driven through a gear box and differential gear. Alternatively, the engine may be situated at the rear of the vehicle and drive the rear wheels through a gear-box and differential gear.

Petrol-engine type, but with hydraulic transmission of power.

Petrol-engine type, with electric transmission. In this case the engine drives an electricity generator, the current from which is fed to an electric motor or motors. In the former example, the motor replaces the gear-box of the first type ; in the latter the motors may be placed on the rear or front wheels.

Automobiles resembling the ordinary kind with which most people are familiar, but employing, in place of the usual petrol-engine, internal combustion engines of the heavy-oil, coal-gas or producer gas types.

Electric vehicles. In this case, a number of electric storage batteries, or accumulators, are carried, current from which drives electric motors connected with the front or rear wheels.

Steam vehicles. Here the petrol-engine is replaced with a steam-engine, and boiler, or other steam generating unit. In design it differs greatly from (1) in dispensing with clutch and gearbox, and in the disposition of its parts. This steam car, however, is now obsolete.

Apart from the types mentioned, there are special vehicles built for travelling over rough country. This type, which includes the caterpillar tractor (examples of which are the Citroen, and the military 'tanks'), and six-wheeled cars such as the Morris, Thorneycroft and Renault cross-desert type. In the former cases the cars provide their own flexible track, and can negotiate rough country.

Mechanical Transmission Car.—This popular type of automobile includes a very wide range, from the very light and economical light car, weighing a few hundredweights, to the luxurious motor-car, employing an eight or twelve cylinder engine capable of developing over 100 horse-power, and weighing from 1½ to 2 tons or more.

At the extreme end of this range is the heavy commercial vehicle, with its 40 to 150 horse-power engine, weighing from 5 to 20 tons, fully laden. Most of these employ the

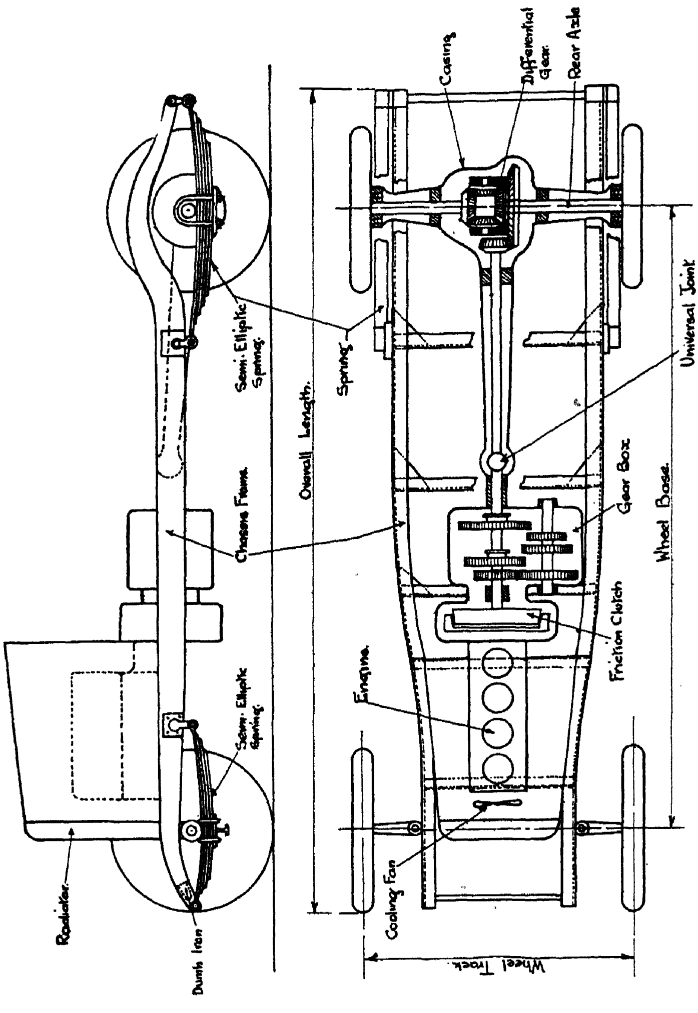


FIG. 1.—Simple Chassis Arrangement.

same general disposition of, and the same mechanica parts; they differ merely in size, design and refinement.

Let us consider briefly, the underlying principles of thi almost universal design of automobile, referring firstly t

the diagrammatic sketch of Fig. 1, and afterwards to the frontispiece and to the other examples of modern cars. The engine is situated at the front end of the car, with the radiator immediately in front, the cooling fan being between the two. The flywheel of the engine, usually forms one member of the *Clutch* unit, the latter name being given to the mechanism for connecting or disconnecting the engine's crankshaft with the *Gearbox* shaft, so that the driver of the car is able to start or to stop the car whilst leaving the engine working. The clutch unit usually consists of two parts, one affixed to the flywheel, or engine, and the other to the gearbox shaft; by means of the driver's clutch foot-pedal they can be made to engage, frictionally, or to disengage the engine drive to the wheels.

The *Gearbox* is the next item in the mechanism for conveying the power to the rear wheels, or, as it is called, the 'Transmission.' The object of the gearbox is to provide means for varying the driving effort, or torque, at the wheels, so that if more effort is required as when starting off from rest, or when climbing steep hills, a suitable gear train may be engaged: when it is required to travel on fairly level roads at normal, or high speeds, quite another gear train is necessary. The gearbox contains three or four different arrangements of gears (known as gear-ratios) for this purpose, as well as a gear-wheel system for enabling the car to be driven rearwards. The former are known as the *Direct*, and the latter as the *Reverse Gears*. In effect then, the gearbox enables the speed of the rear-wheels to be altered (or reversed) relatively to the speed of the engine. Usually the engine works best at a given speed—generally between 2000 and 4000 r.p.m.—and this system of gears enables the engine speed, when giving its greatest power (or fuel economy, in some cases) to be kept constant under most road conditions. It should be mentioned that there is a *Hand Gear Lever* or *Selector* provided, which enables the driver to engage whichever gear he desires.

From the rear end of the gearbox the power is transmitted —usually by a short shaft—to another longer shaft, known as the *Propeller* or *Cardan Shaft*, situated between the gearbox and the rear axle system; this shaft is provided with a bevel gear (or worm) at its rear end, engaging a larger bevel wheel (or worm-wheel) atta

gearing known as the *Differential Gear* to the divided *Rear Axle.*

It will thus be seen that the power is transmitted by

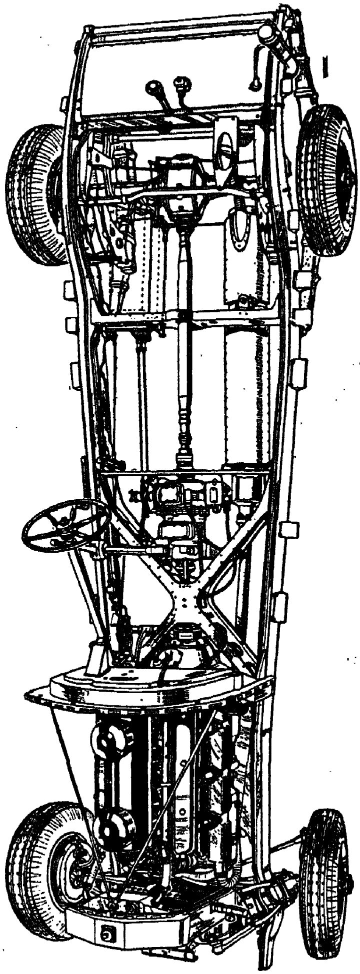


FIG. 2.—The Rolls Royce "Phantom Three" Chassis.

this gearing in a direction at right-angles to the engine and propeller shaft, by the bevel or worm gearing mentioned; this right-angle gearing is known as the *Final Drive.*

At the front end of the propeller shaft is a hinged joint, known as an *Universal Coupling* or *Hooke's Joint* (after the inventor Hooke) which enables the propeller shaft to transmit the power, whilst inclined to the gear-box final-drive shaft. As the gear-box is generally anchored, or fixed to the frame, which also carries the body of the car, whilst the rear-axle is arranged to move vertically up and down under the influence of the rear-wheel springs and the road inequalities, it will be apparent that the propeller shaft must be provided with this universal coupling at the rigid end of the drive, i.e., the gearbox. In some cases, as we shall see, later, another universal coupling is provided at the rear-end of the propeller shaft; frequently, also, there is a sliding movement arranged at one end, to enable the drive to be transmitted whilst the distance between the end of the gear-box final-drive shaft and the axis of the rear-axle system varies under the influence of the springing system; the sliding coupling is sometimes known as a *Sliding Universal*, or in one particular form a *Plunging Coupling*.

Let us revert to the differential gearing, for a moment. We have seen that this is interposed between the final drive and the two separate rear-axles, or *Jack Shafts* as they are termed. This gearing is necessary to enable either jack-shaft (with its rear wheel and tyre) to turn faster or slower than the other, for the purpose of preventing skidding when the car is driven in a curved path, as when turning corners. If no such differential were provided, and a plain single shaft were employed to carry both rear wheels, one wheel would skid on corners, and tyres would therefore wear more rapidly.

The Chassis.—We have now traced the transmission of power from the engine to the rear-wheels, and shown how the practical conditions of road use are provided for in the mechanism. It remains now to show how these various items are attached to a structure to which the actual body of the car is secured. A special light steel frame, known as the *Chassis Frame*, is provided, and the radiator, engine and gear-box, and front wheel steering mechanism are anchored either rigidly, or, as in modern cars, flexibly to it. The front and rear wheels are not attached to this frame directly, but are connected to the *Road Springs*

provided for absorbing road shocks; the other ends of these road springs are attached to the frame, usually, with hinged-joints or *Spring Shackles;* the latter enable the springs to extend in a fore-and-aft direction under the influence of road shocks, or inequalities.

The complete frame, with the power unit, gear-box, propeller shaft, back-axle, rear and front wheels, and the springing system, the steering gear and controls is termed the *Chassis.* The chassis therefore includes the power, transmission, springing and steering mechanisms, and is complete in itself as a propulsion means, or road vehicle. It can be driven and manœuvred by itself, in exactly the same manner as the complete car; indeed it is usual at many motor works to test the chassis on the road before the body is attached.

Other Types of Chassis. For certain reasons we have confined our general survey to the mechanism of the almost universal type of chassis. It should, however, be mentioned that many alternative designs, each differing in detail only, are in use, although all follow the same principle outlined in the preceding pages. Thus there are chassis in which the radiator is behind the engine; others with the engine and gear-box built as a combined unit; some with a supplementary two-speed gear-box between the principal gear-box and the back axle; others with independently driven front or rear wheels and so on.

How the Driving Effort is Transmitted.—We have shown how the power is transmitted from the engine to the rear wheels, but have as yet made no mention of the manner in which the thrust of the rear-wheels is transmitted to the frame of the chassis. The only connection between the back-axle and the frame we have described is through the material of the rear springs. In many cases the springs actually transmit the driving thrust from the rear wheels to the frame; they are designed so as to be strong enough to fulfil this purpose, in addition to the springing action. When the rear springs transmit the thrust, the system is known as the *Hotchkiss* one and is illustrated in Fig. 3 (*b*).

Alternatively, it is general to fit a special thrust-taking member between the back-axle and the frame as shown in Fig. 3 (*c*).

There is another very important point concerning the driving effort, namely, the tendency of the whole car to rotate about the rear-axle when the gears are engaged. It will be apparent that if the rear-axle were held rigidly, the driving pinion of the final drive would tend to rotate around the fixed crown wheel, thus tending to rotate the car about the rear axle, or to lift the front of the car. This effect is termed *Torque Reaction*, and unless special *Torque Members* are provided either the springs or the frame will be severely stressed. Many will no doubt remember the earlier motor-buses, the frames of which deflected noticeably when the clutch was engaged harshly; this effect was due to the cause stated. In the case of chain-driven vehicles it is necessary, for this reason, to provide special torque-rods for both the driving thrust and the torque. The torque-reaction may be taken in one of three different ways, as follows, namely: (1) By the rear springs, specially stiffened; (2) By a *Radius Rod* member as shown in Fig. 3 (*c*), or (3) By means of a spherically-ended propeller shaft casing, with its bearing situated in the frame cross-member as shown in Fig. 3 (*a*); this is by far the most popular method of taking the thrust and the torque.

An alternative arrangement for the third method is to have a forked front end to the propeller shaft casing with pins and bearings in the chassis frame cross member. Further, in order to prevent any sidewise movement stressing the rear springs, it is usual to provide a triangular bracing to the rear-axle and propeller tube, by means of light tension rods.

There are two important points to note in connection with the subject of the driving thrust and torque, as follows, namely:

(*a*) That when the spring portions between the rear axle and the frame take both thrust and torque, *there should be no shackles, or links at the front ends of the springs*, but only bearing pins.

That when radius rods and/or torque members are employed, *the springs should be shackled at both ends;* further, the springs need not necessarily be bolted rigidly to the rear axle casing, but should preferably have trunnion bearings around the axis of the rear axle.

When a radius rod, or *Banjo-Frame* type of torque member is fitted, as shown in Fig. 3 (*c*), the front end

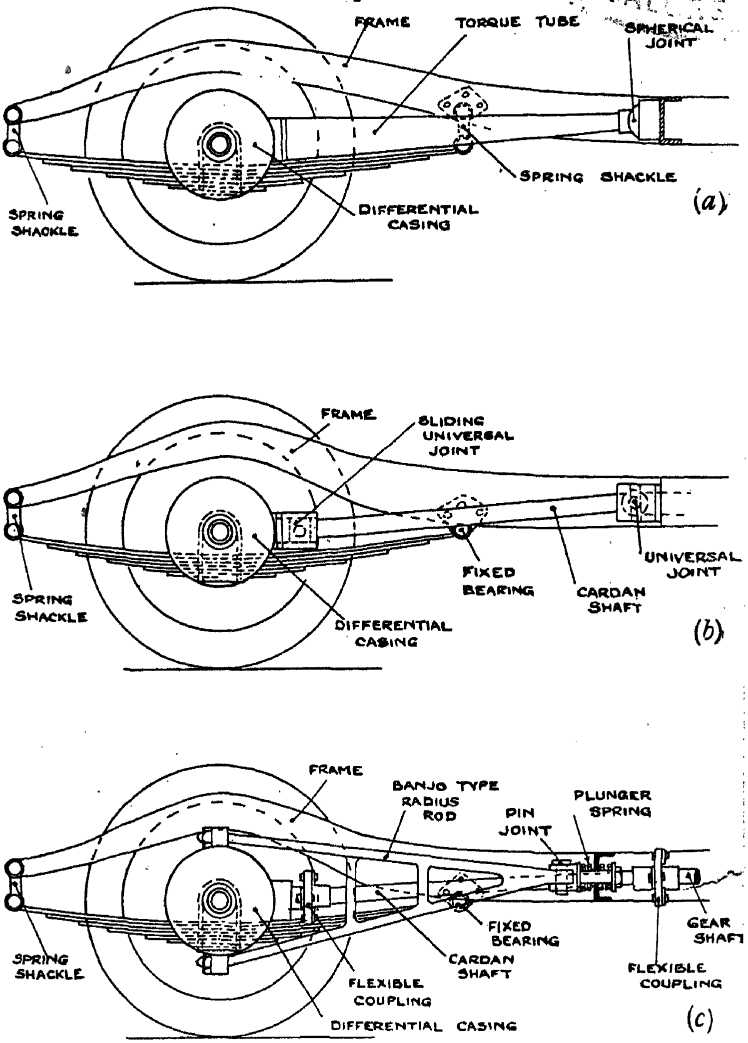


FIG. 3.—Alternative Forms of Final Drives, and Suspensions.

bearing of these, on the frame cross-member, is usually provided with a small amount of fore-and-aft movement,

under the control of a spring, to allow for the vertical movement of rear axle (when the front end of the spring is not shackled), and to give a degree of flexibility to the torque-reaction member.

Mention should here be made of another system of power transmission to the rear wheels, whereby the propeller shaft transmits the drive to the central differential gear assembly of the back axle and the drives for the two rear wheels are taken through cardan, or universally-jointed shafts from the central members to the rear wheels; this method enables the rear wheels to be sprung independently. A fuller account of this method is given later.

The Front Wheels.—A few years ago, with one or two notable exceptions all vehicles were fitted with rigid front axles and passive front wheels, i.e. wheels which were not driven positively but were merely pushed along by the power-driven rear wheels. In recent times, however, attention has been given to the problem of driving the front wheels positively from the engine through a suitable gear-box and differential arrangement; there are several cars in present use embodying this front wheel drive system. In one or two other instances the front and rear wheels are driven positively from the engine so that there is a power drive to all four wheels; further, in one or two cases the rear as well as the front wheels are steered positively. For the present, however, we shall confine our attention to the fixed front axle type of car with power drive to the rear wheels, leaving the subject of independently sprung or driven front wheels until later.

In the case of the simple chassis illustrated in Fig. 1, part of the propulsive effort transmitted from the rear wheels to the frame is transferred to the front wheels through the front springs, the portions of the springs between the front ends of the chassis frame (known as the *Dumb Irons*) and the front axle, actually act *in tension*, and therefore tend to pull the wheels along. If the front springs are of the type so common on light cars, and known as *Quarter-Elliptics* (see Fig. 18), then the chassis frame tends to push the front axle and wheels along, and the springs are *in compression*.

We shall not, at this stage, attempt to describe the

steering mechanism as this may confuse the beginner, but shall merely point out that the front axle is a strong steel cross-member of I-beam section which is attached rigidly to the springs, the steering mechanism being interposed between the front axle and the two front wheel axles; the latter are the short shafts upon which the wheels rotate.

The Importance of Power-Weight Ratio.—In the case of aeroplanes, it was found that the machines which gave the best climbing and level-flying speeds were those which for a given total weight were fitted with engines developing the greatest horse-powers; it should be added that in each case the aeroplane was properly designed and streamlined. It is now fully established that it is neither the power, nor the weight, but the ratio of the power-to-weight which determines the performance of an aeroplane; indeed, the designer can now predict very accurately the maximum speed, the climbing rate and the greatest height (or 'ceiling') which a new aeroplane can attain. Similarly, experience with motor-cars has shown that by keeping the weight of the car down to a minimum, and fitting engines of high horse-power the best performances can be obtained. The *higher the effective brake horse-power* of the engine, and the *lower the total weight* of the car (complete with occupants) the *better* will be its *hill-climbing qualities* on a given gear-ratio, the *higher* its *maximum speed* and the *better* its *acceleration*, that is to say its ability to increase speed from a standstill, or a lower speed. Moreover, a high power-to-weight ratio, in a well-designed car, also results in a *low fuel consumption.*

In connection with the power-to-weight ratio most of the progress which has been made in motor car performance has been due to a progressive increase in this ratio. This has been brought about by the use of lighter materials and improved methods of chassis and body construction, whereby the total weight of the car has been reduced considerably. In addition, the power output of the engine has been increased by such means as the use of aluminium and magnesium alloys, increased compression ratios (due to the use of aluminium alloy pistons and cylinder heads), fuels of higher octane values, improved piston, piston-ring and cylinder design, combustion

chamber modifications, better lubrication systems, improved engine balance and other contributory factors.

Whilst many years ago the ratio of the b.h.p. to the car weight (in cwts.) was usually of the order 0.3 to 0.9, to-day this ratio has been increased so that for modern cars it lies between 1.8 and 3.8, averaging about 2.2. Most of the high performance touring cars have values lying between about 2.5 and 3.8. It is significant that, given a reasonable body design in regard to wind resistance, the power-to-weight ratio determines the road performance, of any car, irrespective of its actual rating so that the same performance, i.e. acceleration and hill climbing ability could theoretically be obtained from a 10 h.p. car as from one of, say, 25 h.p. rating, if the ratio in question were the same.

From the point of view of maximum speed only, whilst the power-to-weight factor is of primary importance, questions of minimum body resistance are also involved, since the air resistance of an automobile body increases approximately as the square, whilst the power absorbed varies as the cube of the speed.

Engine Capacity and Car Weight.—It was the rule, in pre-war days, to allow for all motor vehicle types an engine capacity of 100 cubic centimetres per cwt. of car weight. This rule is still followed, to some extent, to-day, but modern engines give considerably higher outputs—in many cases from two to three times the power for the same cylinder capacity, so that the actual power-to-weight ratios are considerably better.

An examination of the various car weights and corresponding cylinder capacities in the instances of recent makes reveals the fact that for mass-production models of 7 h.p. to 10 h.p. rating the allowance works out at 60 to 70 c.c. per cwt., reckoned on the empty car's weight.

For cars with engines of 12 to 15 h.p. the corresponding figures are from 70 to 80 c.c. per cwt. Cars of about 20 h.p. fitted with ordinary five-seater bodies give from 80 to 100 c.c. per cwt.

The higher powered cars such as the Ford Vee-eight and typical American cars of 25 to 35 h.p. rating give values varying from 125 to 150 c.c. per cwt., whilst specially

built road sports model cars, with light touring bodies give from 150 to 200 c.c. per cwt.

It is of interest to note that the standard model touring car, having the best acceleration at the time of this edition, gave 152 c.c. per cwt.

Performance Values for Special Cars. Whilst the preceding methods of assessing the cylinder capacity, power output and weight characteristics of cars give fairly accurate results for non-supercharged cars of normal design it will be found that it is not possible to treat special racing or record-attempt cars on the basis of cylinder capacity to weight. Thus, if we consider two examples taken at the extremes in racing cars, namely the 1,086 c.c. M.G. car with which Major Gardner raised the existing 148·8 m.p.h. road record to 187·616 m.p.h. for the flying mile in Germany, and the 73,164 c.c. Thunderbolt car of G. E. T. Eyston, which established a speed of 312·2 m.p.h. in 1937, it will be found that whereas the former gave 72·4 c.c. per cwt., the latter gave no less than 522·6 c.c. per cwt.

If, however, one considers the actual power-to-weight values, these work out at 11·3 b.h.p. per cwt. for the M.G. car and 25·7 b.h.p. per cwt. for the Thunderbolt car. These values are directly comparative, and correspond with the relative performances of the two cars. It is also of interest to note that the M.G. engine developed 170 b.h.p. at 7,000 r.p.m., giving an output of about 152 b.h.p. per litre (1000 c.c.), and the two twelve cylinder Rolls Royce engines of the Thunderbolt of 73,164 c.c. total capacity developed 3,600 b.h.p., which works out at about 49·3 b.h.p. per litre; the larger car, however, was much lighter per h.p. than the smaller one.

Fuel and Oil Consumptions.—The fuel and oil consumptions do not depend upon the power-to-weight ratio, but upon the total power or total weight. The higher the power, or weight, the greater will be the consumptions. As a result of a large number of tests it has been found that for modern cars, of reputed design, there is a definite relation between the petrol consumption and horse-power.

For the average saloon car of modern design the following are representative petrol consumptions:

TABLE No. I.

H.P. (R.A.C.) .	7	10	12	14	16	18	20	24
Petrol Consumption (miles per gallon) ...	42	35	31	26	23	20	18	16

For lighter cars of the same rated horse-power, such as two-seaters, the fuel consumptions will be lower and the M.P.G. higher than the values given in Table No. 1.

In general, for a car of given horse-power rating the fuel consumption will increase in proportion to the weight of the loaded car, at any given road speed.

With modern carburettors which are arranged to run normally on rather weaker mixtures than those for full power, except when the throttle is full open—when a richer mixture is provided, the petrol consumptions per mile have been reduced. Thus, in the case of three cars of the same make, of 10, 12 and 14 H.P. (R.A.C.) rating the average petrol consumptions were 40, 35 and 30 M.P.G., respectively.

The above consumptions represent the averages over ordinary give and take main roads, in dry weather and with proper carburation. It may be added that a modern petrol engine will show a full-load petrol consumption of about 0·6 pints per b.h.p. per hour. Most car engines, it should be added, develop at least three times the horse-power given by the R.A.C. rating.

In regard to the oil consumption of modern car engines this depends largely upon the design of the engine and upon its size. With well designed lubricating systems having efficient oil cleaners and with the use of suitable grades of oil the oil consumption works out at 1,000 to 2,000 miles per gallon, dependent upon the size of engine, the smaller one giving the higher mileage value.

A more accurate figure for modern engines is 0·025 to 0.030 lb. of oil consumed per hour per brake horse power for new condition engines.

The Road Performances of Cars. As a result of progressive improvement in the design of cars and in the methods and materials of construction the performances of ordinary motor cars have benefited in the matter of acceleration,

maximum road speeds and hill-climbing qualities, so that the smaller mass-production cars of to-day have better performances than the luxury cars of a few years ago; indeed, the only limiting factors now appear to be the conditions of the roads themselves, in this country.

The smallest class of car is the 7 to 8 h.p. rated one. These cars have wheel-tracks of about 3 ft. 7 ins. to 3 ft. 10 ins. and wheel-bases of 6 ft. 9 ins. to 7 ft. 6 ins. They weigh from 12 to 15 cwt., empty.

The maximum attainable level road speeds on top gear for these mass-production cars range from about 55 m.p.h. to 60 m.p.h., according to the make.

The best acceleration values from 0 to 50 m.p.h. through the gears are from 30 to 40 seconds and the weight per engine c.c. from 1·6 to 1·9 lb.

Commercial cars of the 10 h.p. class have wheel tracks of 3 ft. 10 ins. to 4 ft. 2 ins. and wheel-bases of 7 ft. 6 ins. to 8 ft. 6 ins.; the weights vary from about 15 to 20 cwt.

The maximum road speeds lie between 55 and 65 m.p.h., as a rule. The acceleration times from 0 to 50 m.p.h. vary from about 25 to 35 sec. and the weight per c.c. from 1·5 to 2·5 lb.

Considering next the 12 to 14 h.p. class of car, the wheel-tracks of these usually lie between 4 ft. 3 in. and 4 ft. 10 in. and the wheel-bases, 8 ft. 6 in. to 9 ft. 6 in. The weights range from about 20 to 23 cwt.

The maximum road speeds usually achieved under favourable conditions are from 65 to 75 m.p.h. according to the make.

The acceleration times from 0 to 50 m.p.h. generally lie between 20 and 30 seconds.

Turning next to the 20 to 25 h.p. class of English car, this is made in range of chassis sizes with wheel tracks of 4 ft. 8 in. to 5 ft. 0 in. and wheel-bases of 9 ft. 6 in. to 12 ft. 0 in. The car weights vary from 25 to 38 cwt. according to the size and type of body fitted.

The maximum road speeds are generally higher than those of the smaller cars, except for certain sports models; these speeds usually lie between 70 and 85 m.p.h., a good average being 75 m.p.h. for a 25 h.p. car fitted with saloon body. The acceleration times from 0 to 50 m.p.h. lie between 15 and 20 secs.

In the 25 to 50 h.p. rated class, there are several makes

of car that will achieve maximum speeds of 85 to 100 m.p.h., the latter speeds usually being reached by supercharged cars, or specially built sports models.

The average American car of 25 to 30 h.p. rating has a maximum road speed of about 80 to 85 m.p.h. and an acceleration time, from 0 to 50 m.p.h. of 14 to 17 seconds. The weight of car per c.c. engine capacity lies between 0·85 and 1·00 lb.

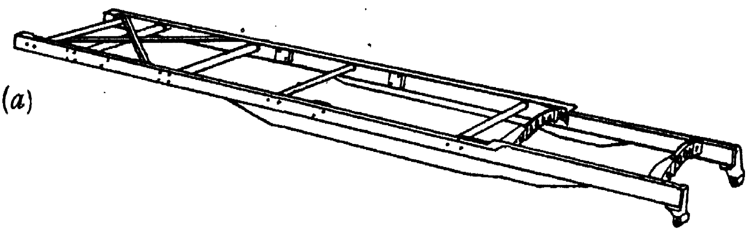
CHAPTER II

THE FRAME AND SUSPENSION SYSTEM

Having described briefly the principal components of the mechanism, it now remains to show how these are mounted, or fixed, relatively to each other, and also to consider how the bodywork is fitted.

The engine, gearbox, steering gearbox and the springs forming the suspension system are all mounted upon a special metal structure known as the *Frame*. In the early days of motor-cars, these frames were constructed, in rectangular form, of ash, oak or hickory, the members being joined by means of steel flitch plates bolted to them. The later practice was to employ steel pressings of channel

(*a*)



(*b*)

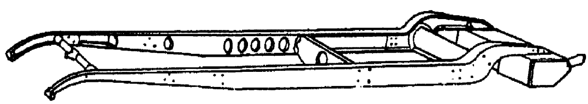


FIG. 4.—(*a*) Commercial Vehicle Frame.
(*b*) Earlier Type of Car Frame.

section, to form the side-members, that is the two members running fore-and-aft, and to connect these side-members by means of other channel sections, or tubes, so as to form a rigid but light framework. The cross-members were also used to mount the chassis components upon. These lateral members were usually riveted to the side-members, special enlarged flanges being provided for this purpose as shown in the frontispiece. At the front ends, and

usually also at the rear ends the side-members tapered in depth, and the front-members were also closer together in plan view; this tapering is in conformity with the

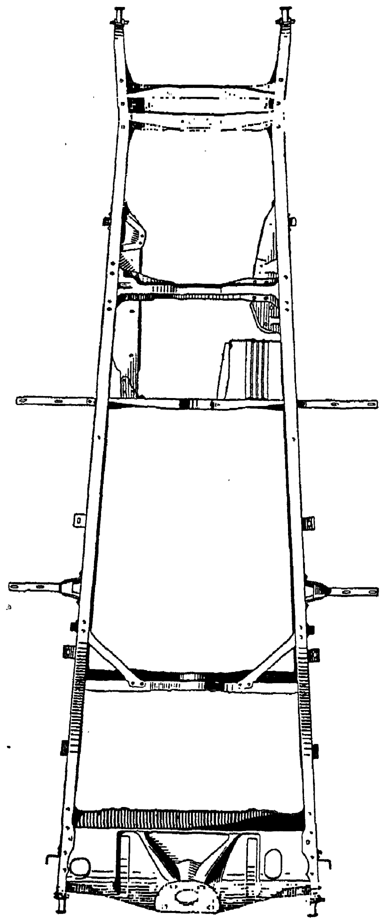


FIG. 5.—Plan View of Earlier Type 20 h.p. Car Frame for rigid engine mounting.

requirements of metal beams of uniform strength for minimum weight.

At the front ends, metal forgings were let in and riveted; these formed the bearings for the front spring shackles, and are known as the *Dumb Irons*; the same term is applied

also to the rear-members, when fitted. It was the practice also, to bolt special brackets, in the shape of steel stampings or drop-forgings, to the sides (or webs) of the frame, to house the inner ends of the springs, or in the case of rear cantilever springs, the trunnions and front ends.

In regard to the rear end of the frame it was usual to provide one or more cross strut members of channel or tubular section in order to take the rear spring forces tending to distort the frame and to provide a certain amount of rigidity in the lateral sense. It was usual, also, to arch the frames towards the rear end, in order to provide sufficient clearance for the vertical movement of the back axle under its springing action.

The frame as viewed in plan is invariably tapered inwards towards the front of the car in order to enable the front wheels—which are about the same distance apart as the rear ones—to be moved under the influence of the steering mechanism without touching the frame.

In order to keep the bodywork as low as possible, it follows that the frame should be arranged with this object in view; for this reason the frame can be kept low between the wheels and upswept over the front and rear axles, respectively; alternatively the front end of the frame can be down-swept below the front axle.

As will be explained later, the front-drive car enables a much lower body position to be obtained than the normal rear-driven type.

More Recent Design Tendencies. The type of frame described was used over a long period and gave satisfactory results. Further, as it was of simple rectangular design it was not difficult to repair in cases of minor accidental damages; the modern cross-braced frame is much more difficult to repair.

In earlier designs of chassis the engine and radiator were rigidly secured to the frame, the former acting as a kind of frame stiffening member, whereby the engine crankcase was submitted to road shock stresses.

In later designs the frame is stiffened considerably by the use of stronger frame sections of the box or welded stiffener pattern and with the aid of diagonal struts riveted or welded across the frame side members. (Fig. 6.)

The engine is now mounted flexibly, by means of rubber

blocks, to the frame at three or four places in order to insulate the frame and the car body which is attached to the latter, from engine vibration, and to lessen the effect of road shocks upon the engine.

Under these circumstances the front portion of the frame

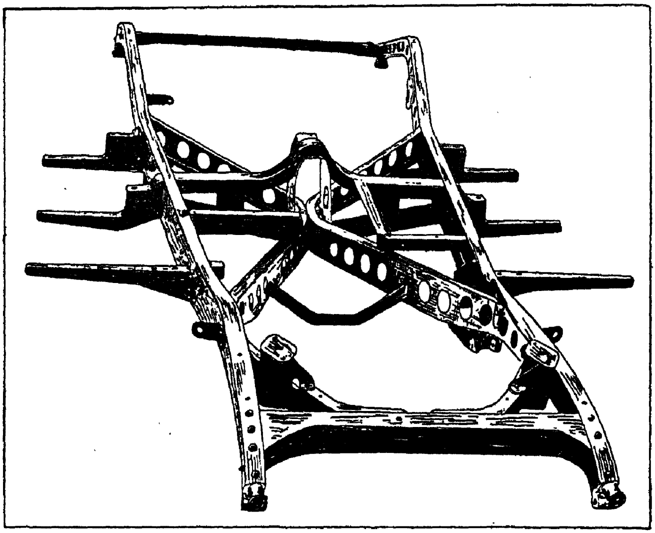


FIG. 6.—Wolseley Chassis Frame.

is now made of sufficient strength and rigidity to take any normal road shocks without distortion, the engine being freed from chassis stress effects.

There are various alternative methods of mounting the engine flexibly, as follows, namely, (1) At two front and two rear places; (2) At one central position on the front transverse frame member and at two rear places and (3) At two forward places, one on either side of the engine and at one rear place corresponding to the gear-box end.

The so-called three-point suspension method is used with the object of preventing any serious frame distortion from affecting the engine, i.e. subjecting it to stress.

In order to carry out this suspension method one anchoring should be a ball and socket joint and the other two parallel pin joints. With the flexible rubber block engine mountings, however, an appreciable amount of

frame distortion may occur without affecting the engine unit.

An important point to remember in connection with flexible engine supports is that all externally connected

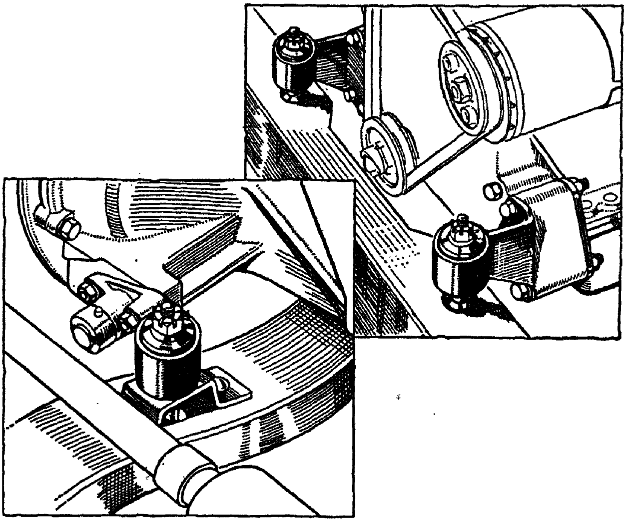


FIG. 7.—Flexible Rubber Supports between Engine and Chassis Frame. (Left) Rear End. (Right) Front End of Engine.

tems, such as the exhaust pipe, petrol pipe to rear tank and the electric leads should also be flexibly attached at the engine or chassis frame ends.

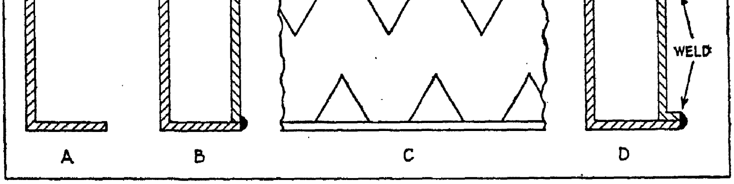


FIG. 8.—Chassis Frame Sections.

Frame Stiffening Methods. In place of the usual channel sections (*A*, Fig. 8) for the side and certain cross members

it is now the custom to employ box sections of the welded or built-up type; typical examples are shown at B, C and D in Fig. 8. These sections provide greater vertical and lateral strength besides being much stronger in torsion.

Examples of modern frame design are shown in Figs. 9 and 10 in the case of the Austin larger cars. The former illustration indicates the method of stiffening the frame

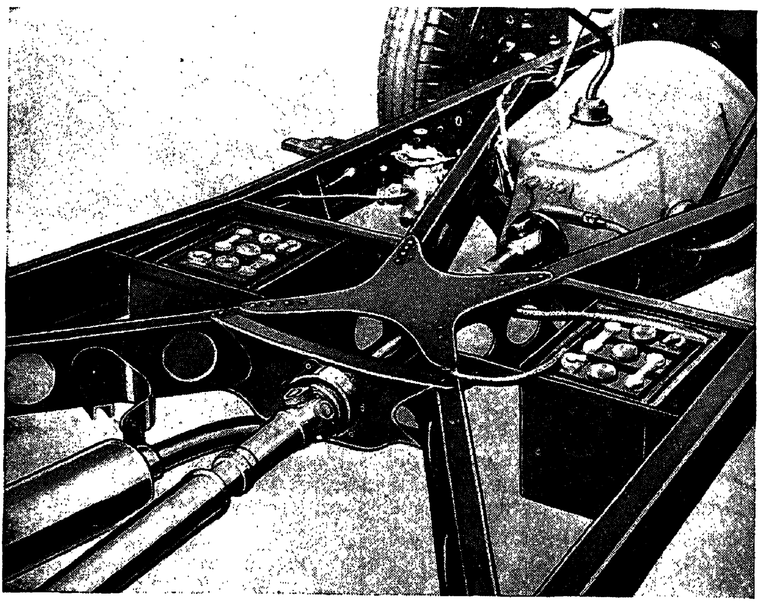


FIG. 9.—Austin Twenty-Eight Frame Stiffening Members.

laterally by means of diagonal members the junction of which forms the rigid housing for a rubber insulated bearing for the rear end of a short propeller shaft between the gear box and the front end of the propeller shaft universal joint. The engine and gear box are mounted on flexible rubber members so that the transmission is insulated from the frame against vibration and shocks.

Fig. 10 shows the rear end of the chassis of the Austin Twenty-Eight car and reveals how the cross-bracing member shown in Fig. 9 extends along the rear end of the frame so as to form a rigid box section. The inner vertical sides of this section are lightened by means of the holes shown.

The rear end of the frame is upswept to clear the back axle assembly. Other features shown include the Girling

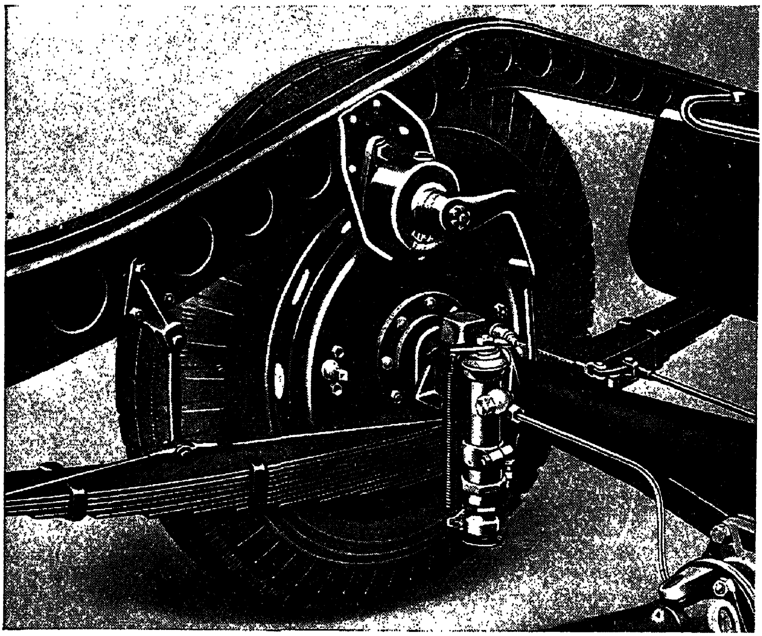


FIG. 10.—Rear End of Austin Twenty-Eight Chassis.

brake actuating mechanism*, hydraulic shock absorbers and positively lubricated low-periodicity leaf spring.

A typical design of frame for a high-powered American

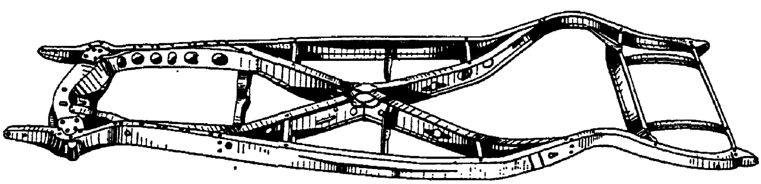


FIG. 11.—The Oldsmobile Frame for Independent Front Wheel Springing.

make of car, namely, the Oldsmobile, is illustrated in Fig. 11. It consists of heavy section side members and sub-bars, a

*Described on page 274.

heavy " X " cross member and four straight cross members. The side members are of channel section and the " X " cross member of " I " beam form. It will be observed that the " X " cross member extends from the rear engine mounting to the upswept portion of the frame over the rear axle. The member in question has bars which are joined at the centre by heavy gusset plates and braced to each frame side rail at three points. The sub-bars are added at the front junction of the " X " member and the frame side rails; this forms a box section of the frame side rail flange and the sub bars.

The front cross member is a complete box section riveted to the frame side rails and supported, additionally, by the forward end of the frame sub-bars which are riveted to it.

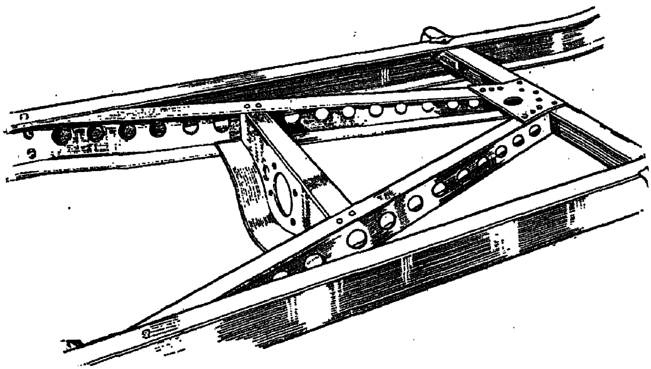


FIG. 12.—Triangular Stiffening Member.

This extremely rigid design of the front end of the frame is necessary in view of the fact that the car in question has *independent front wheel springing.*

The frame side members, as previously stated, are usually braced diagonally as viewed in plan form by a cruciform member welded or riveted in position. In some cases a triangular stiffening member is employed, as shown in Fig. 12; here, the front cross member forms the centre support for the propeller shaft.

Frame Dimensions and Materials.—In the case of light cars, the overall dimensions of the frames average 8 to 11 ft. in length (including the springs) by 2 to 3 ft. in width. The frame usually tapers from about 3 ft. or

3 ft. 6 in. at the rear to 1 ft. 9 in. to 2 ft. 6 in. at the front, as viewed in plan. The maximum cross-section of the side-members is usually towards the rear end of the frame and is usually about $2\frac{3}{4}$ to $3\frac{1}{4}$ in. deep, with flanges of $1\frac{1}{4}$ to $1\frac{1}{2}$ in., the steel being $\frac{1}{8}$ to $\frac{3}{16}$ in. thick. In the case of the frames of large cars (20 to 40 h.p.), the overall dimensions are of the order, 12 to 16 ft., and the width between the side-members from 3 ft. to 3 ft. 9 in. A typical maximum cross-section of the side-member channel would be from 4 to 6 in. deep, 2 to 3 in. wide, and thickness from $\frac{5}{32}$ in. to $\frac{1}{4}$ in.

The material employed for pressed frames is usually a low nickel steel containing from 2 to 3 per cent. nickel, and having a breaking strength in tension of 35 to 40 tons per sq. in. Frequently, pressed steel frames are made from a high grade of mild steel of breaking strength 30 to 40 tons per sq. in., and carbon content of 0·3 per cent.

Combined Chassis and Body. Hitherto, it has been usual to design the chassis and frame first and the body afterwards, thus making the design of the latter dependent upon the former items. In certain recent instances, notably the Vauxhall Ten, the body has been designed first and the chassis afterwards, but with the body framework arranged as a kind of deep box girder to take the place of the usual frame. The whole depth of the body is here utilized for taking stresses hitherto taken by the frame alone.

In addition, various parts of the body are reinforced; the floor by the propeller shaft tunnel; the doors by their double construction and the scuttle by the steel bridge which carries the facia; and so on.

By means of suitable ribs or grooves pressed into the sheet metal sections the strength of the latter against bending has been increased materially.

In the Vauxhall example there is a separate front end section, taking the front suspension and the engine, which is attached to the front portion of the body.

This method of construction utilizes the welding process for joining the various members forming the combination body and frame; in certain cases a combination of welding and riveting is employed. Whilst the combination body and frame gives at least equal strength it is definitely

lighter than the separate body and chassis arrangement.

The only drawback to the combined body and frame is that in the event of accidents it is more difficult to

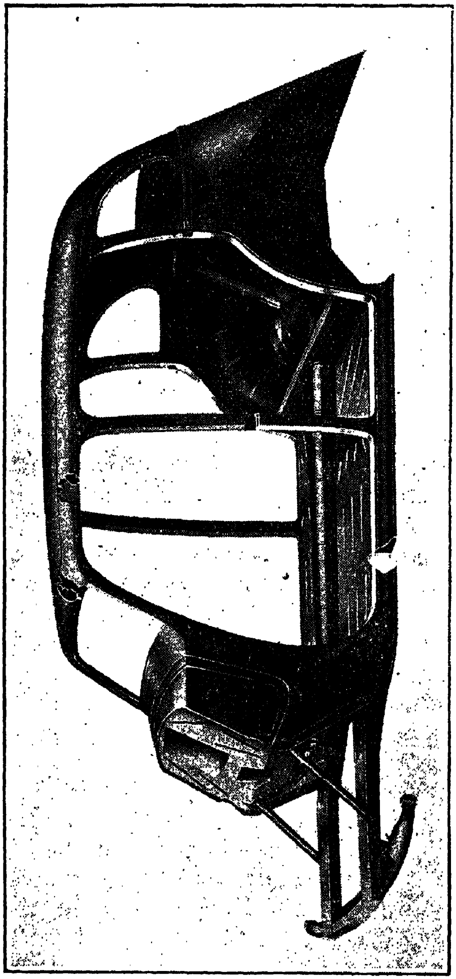


FIG. 13.—Vauxhall Combined Chassis and Body Construction.

repair; in serious cases it may be necessary to scrap the combination unit entirely and substitute a new one. The same objection applies to some extent to the modern welded box-section cruciform frame.

Aluminium Alloy Frame. The use of cast aluminium alloy members bolted together to form the chassis frame and sometimes part of the body as well is an alternative to the sheet metal frame and body. An interesting example

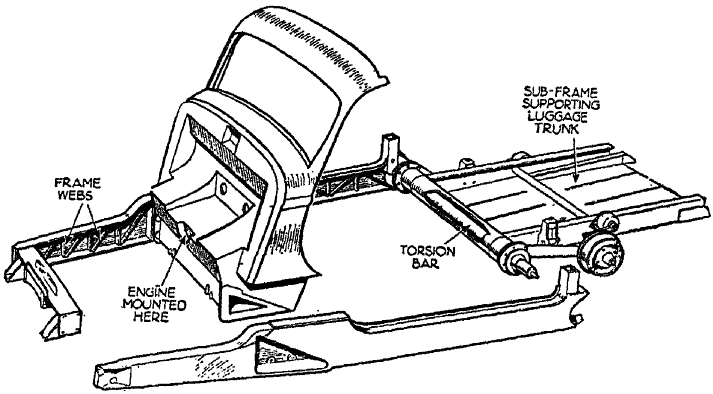


FIG. 14.—Built-up Aluminium Alloy Frame of the Amilcar Car.

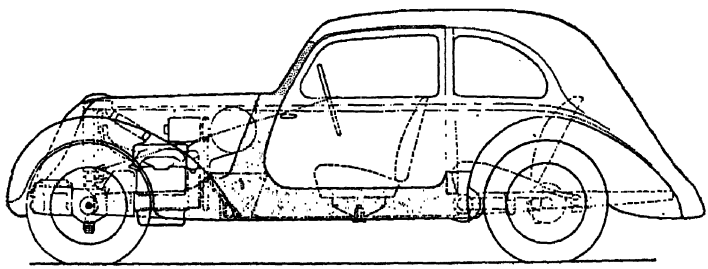


FIG. 15.—The Amilcar Car, showing Alpax Frame.

of this construction is that of the 10 h.p. Hotchkiss Amilcar*, illustrated in Figs. 14 and 15.

The aluminium alloy used was Alpax and the system employed included two side members with internal ribs; a complete cast Alpax dashboard and windscreen unit; a forward cross-frame member and a rear tubular cross-frame one also. The five units mentioned were bolted together, thus giving a rigid construction, such that it was possible to jack up the chassis at any one place without any measurable distortion occurring at any other part;

* *The Autocar*, July 29th, 1938.

indeed, so rigid was the combination that after miles gruelling journey from Europe to Persia the was silent and the doors shut as satisfactorily as when new.

The rear part of the frame was provided with a tail member made of two light U-section steel frame members

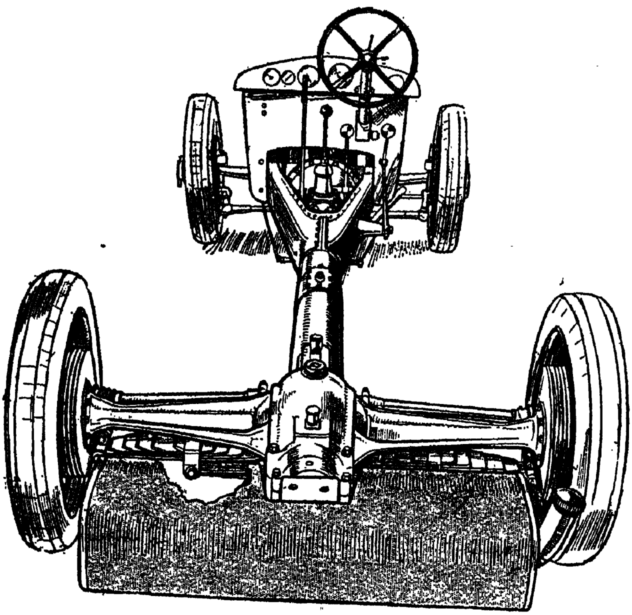


FIG. 16.—Austro-Daimler Tubular Chassis.

joined by means of a sheet steel floor; this unit housed the petrol tank and shock absorbers. The rear wheels of this front-drive car were independently sprung, each rear wheel being carried on the rear end of a radius arm, the other end of which was rigidly secured to one end of a torsion tube; the other end of the latter was secured to the frame, thus giving torsion-bar springing—to which reference is made later in this book.

In regard to the weights of the members, the total weight of the Alpax items, shown in Fig. 14, was 131 lbs. The weight of the complete chassis and body was about 1¾ cwt. less than for one of the normal construction.

A Tubular Chassis.—The ordinary flat frame type of chassis has to be made sufficiently strong to resist torsion, so that the principal members, viz., the engine and

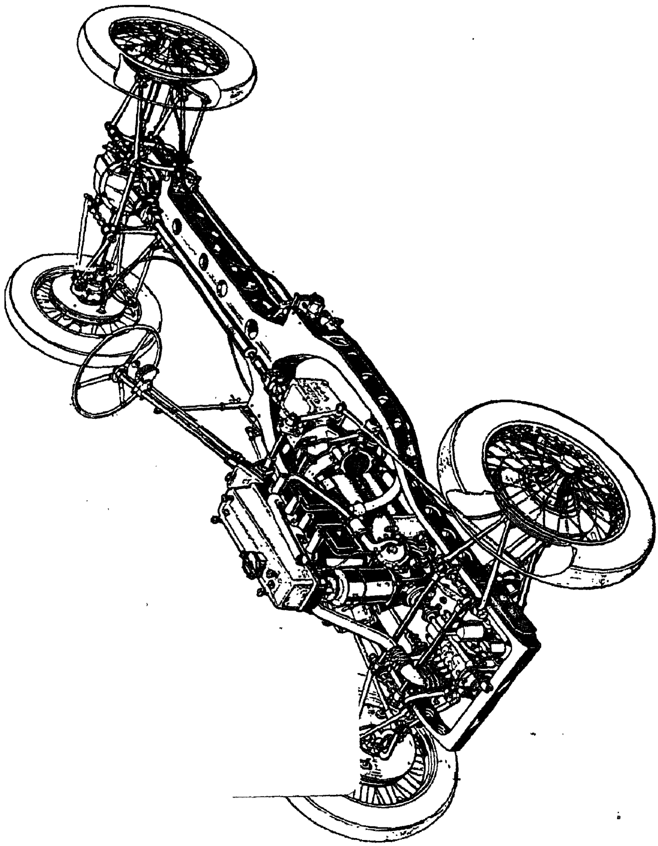


FIG. 17.—M.G. Racing Car Chassis.

transmission parts, are not stressed in any way by road shocks.

The flat frame is by no means the best form to resist torsion, however, and for this reason certain firms, *e.g.*, the Austro-Daimler and German Tatra, adopted a tubular chassis of the type shown in Fig. 16.

In this case the engine, gearbox and differential unit are accurately aligned in a central tubular member, forked at the front end to take the engine assembly. The front axle

is also carefully positioned to this frame. The rear wheels are sprung separately and each is driven by a cardan-shaft from the differential gear. Transverse flat laminated springs are used in this case. There are two cantilever springs and a common connecting leaf spring situated below the differential casing. This arrangement gives great strength in torsion, and bending—in the longitudinal sense—although it is not so convenient as the flat frame for body mounting. Fig. 17 shows another example of the " backbone " method of frame construction, namely the M.G. racing car chassis. This has a divided frame at the forward end and single backbone member at the rear. An interesting feature of the chassis is the use of torsion bar springing on all of its wheels; this method is explained on page 59.

Another example of the " backbone " method of frame

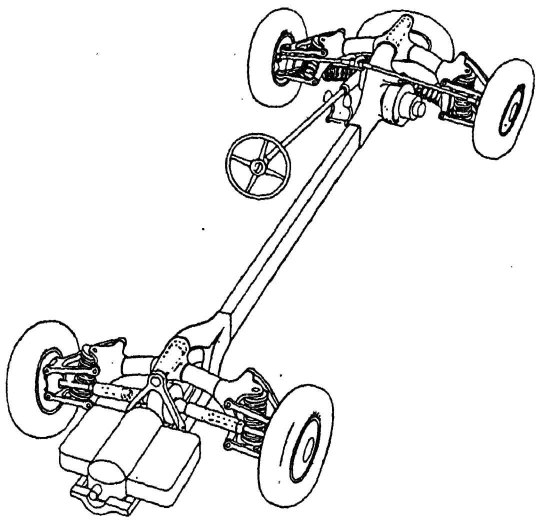


FIG. 18.—" Backbone " Chassis for Rear Engine Drive and Independent Wheel Springing.

construction, which is designed for use with a rear-engine type of chassis is shown in Fig. 18. This system employs a rectangular enclosed sheet metal box section " backbone," with strong arched tubular supports at the front and rear for the independent helical spring suspension system.

The rear portion of the frame is forked to allow for the location of the gearbox. The engine shown is of the opposed cylinder, flat or "pancake" form with direct drive to the gearbox in front of it. From the latter power is transmitted to each of the rear wheels by means of cardan shafts in a similar manner to that used in the Citroen front driven car.

Springing of the Car.—If the front and rear wheel axles were allowed to run in bearings fixed rigidly to the frame, the result would be extremely uncomfortable, the maximum speed of the car would be very limited, and the engine and transmission, as well as the bodywork, would be subjected to severe stresses, which in time would no doubt result in the fracture, or breakdown of one or other of the working parts.

It has become recognised, as a result of long experience, that all types of vehicles used for locomotion, including railway trains, motor vehicles, horse-drawn vehicles, pedal cycles and even children's prams, must be provided with some means of insulating the wheels and axles, from the rest of the vehicle, so that the road, or rail, shocks received by the wheels when travelling over uneven ground, will not be transmitted appreciably to the other parts. The axles of railway carriages run in gunmetal axle-boxes which can slide vertically in guides (known as 'horn plates') in the carriage frames; stiff springs bear down on the tops of these boxes and absorb most of the rail shocks; i.e. spring-insulate the carriage frames from the wheels and axles. The familiar leaf-springs of horse-drawn vehicles serve also for the same purpose. Similarly, the pneumatic tyres and the spring saddles of pedal cycles afford a fair degree of insulation from road shocks.

The object of the springing, or as it is termed, the *Suspension System* then is: (1) To protect the occupants from road shocks; (2) To reduce the stresses due to road shocks on the mechanism of the car, and (3) To maintain the body on an even keel when travelling over rough ground, or when turning, so that any rolling, pitching or vertical movement tendency is minimised. The ideal suspension system would be that which allowed the road wheels to travel over rough uneven ground at any speed, whilst maintaining the body perfectly level; the wheels

would therefore move up and down relatively to the body.

The Principles of Motor Springing.—Before outlining the usual methods of springing cars it should be mentioned that in the earlier days of motor cars, members of the engine and transmission, and in some cases the frames themselves, were apt to fracture through 'fatigue' of the metal under the rapidly alternating stresses caused by road shocks, so that the importance of protecting these parts will be appreciated.

The important principles underlying the satisfactory springing of motor vehicles are firstly, *the reduction, to a minimum, of weight of the wheels and other parts receiving the road shocks;* this is usually termed 'reduction of unsprung weight.' Secondly, *the reduction of rolling or pitching of the body, to a minimum,* by suitable design and attachment of the springs. It is usual to mount the body-frame on the springing system at four points—generally at the corners of the rectangle formed by the frame members. Thirdly, it has become recognised that it *is not yet possible to absorb satisfactorily the larger and also the smaller road impacts* with one springing device, so that auxiliary attachments, or subsidiary members of the main springs are provided to look after the minor shocks; these are termed *shock absorbers.*

How Cars are Sprung.—Most cars depend for their main springing upon the use of laminated steel, or leaf springs, and endeavour to assist this main suspension by the use of pneumatic tyres, auxiliary mechanical shock-absorbers, and the cushioning effect of the upholstery.

The more common system is the combination of laminated steel springs enclosed in grease-retaining leather covers, large-section low-pressure or 'balloon' tyres, shock absorbers for each wheel, and well-sprung upholstery of spiral-spring and horse-hair, or the pneumatic cushion type.

In certain modern designs of chassis the coil spring has been used in place of the laminated type, more particularly for independently sprung front and rear wheels; in this way a saving in length or space has been achieved.

Another type of springing that may here be mentioned

is the *torsion bar* system, in which a heat-treated alloy steel bar is arranged to act as a kind of torsion spring to absorb the major road shocks. More recently an appreciable amount of attention has been given to chassis in which the two *front wheels are sprung independently* of each other—a method having certain advantages over the usual solid front axle one. The subject of independently sprung rear wheels has also been studied and certain designs of car based upon it are marketed.

The reduction in the unsprung weight given by these systems represents a further advantage over the solid front and rear axle springing methods.

The Laminated Spring.—Fig. 19 illustrates the commonest type of laminated leaf spring. It consists of

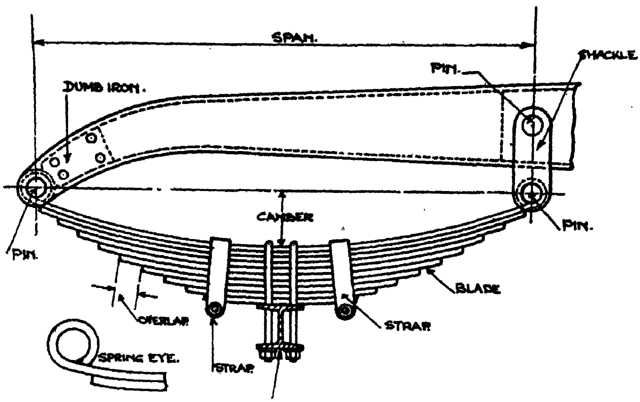


FIG. 19.—Components of a Semi-Elliptic Front Axle Spring. (Not to Scale.)

a number of steel strips or blades of varying length, mounted together in the manner shown, so as to form, virtually, a steel beam thickest at the centre and thinnest at the ends. As the engineering reader well knows, this gives a steel beam of uniform strength at all parts; this is the lightest possible beam for its strength. The object in using a number of blades is to allow more vertical movement, and to introduce a number of frictional surfaces (between the blades) in order to damp down the oscillations of the whole spring. When the axle is deflected upward

by a road bump, the spring having a period, or time of vibration of its own—just like any other spring, helical or flat—will be set in oscillation ; if this motion is allowed to continue the car will tend to bounce along, since the spring is attached to the frame at its ends. Owing, however, to the relative sliding of the individual blades and to the pressure between them as they deflect, a considerable amount of resistance, or damping action is experienced, which is effective in stopping the oscillations of the spring as a whole. In this respect it has been found that earlier pattern springs designed to work in exposed

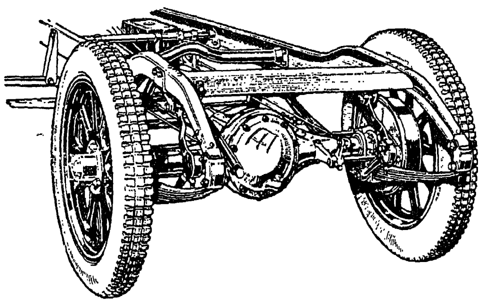


FIG. 20.—Rear View of Earlier Small Car Chassis, showing Frame and Semi-Elliptic Springs.

positions, will tend to promote bouncing if enclosed in grease-retaining spring covers, or gaiters. On the other hand, the spring designed for use with oil or grease lubricating covers, is somewhat stiffer and when lubricated gives about the correct degree of damping.

As each spring has its own period of vibration, it is usual not only to make the front and rear springs of different period, to prevent dangerous resonance effects, but also in the case of semi-elliptic springs to make the two portions of each spring of unequal length so that each has its own period.

Referring to Fig. 19 it will be seen that the two ends of the spring are provided with holes, or eyes. The front end is connected by a pin-joint, or hinge, to the front end of the chassis frame dumb iron. The rear end is hinged to a link, or *Spring Shackle*, which itself is hinged to a bracket on the frame. It is essential to provide at least a shackle with this type of spring, to allow for the

longitudinal extension, or flattening out of the spring when loaded; the shackle rocks about the frame bracket. The blades are bolted rigidly to the front (or rear) axle on special pads or brackets, and are provided with U-shaped metal straps to hold them against lateral sliding.

Spring Leaf Liners. In order to prevent leaf rusting and squeaking, thin liners of special material such as zinc or other softer metal than the steel of the blades, or alternatively brake lining material, are inserted between the blades. When zinc interleaves are employed they serve also to maintain the frictional coefficient between the blades at a constant value, so that the damping effect of this blade friction remains uniform over a long period of service.

Ordinary leaf springs which tend to rust and become noisy in action are now sprayed with a penetrating oil once every 2,000 miles or so.

Fig. 21 shows a section of the rear leaf spring unit

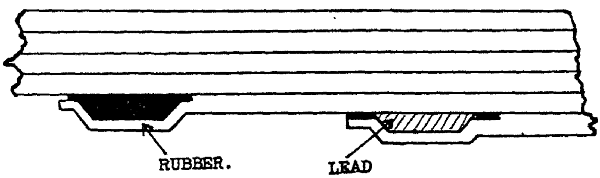


FIG. 21.—Rubber and Lead Spring Inserts.

employed on the more recent Packard cars. The ends of all leaves, with the exception of the bottom three, are provided with moulded rubber inserts. The last three blades have high antimony-lead buttons housed in a special brass retainer with a moulded rubber seal to enclose the lubricant. The object of the lead inserts is to provide a certain amount of damping action for the low frequency vibrations and also to give full-load passenger loading control.

Spring Materials.—The best shock-resisting steels only are used for spring blades. These steels possess high tensile and fatigue strength, maximum resilience, and, of course, minimum weight. The following are suitable steels (1) *Carbon Steels* (0·5 to 1·0 per cent. carbon), heat-treated

to give a tensile breaking strength of 75 to 90 tons per sq. in; (2) *Chrome Vanadium Steels* (0·5 to 1·0 per cent., carbon, about 1 per cent. chromium and 0·2 per cent. vanadium), heat treated to give 90 to 100 tons per sq. inch tensile breaking strength; (3) *Silico Manganese Steels* (0·5 to 1·0 per cent. carbon, 1·9 to 2·2 per cent. silicon, and 0·5 to 0·8 per cent. manganese), heat-treated to give a tensile breaking strength of 85 to 95 tons per sq. in. It should be mentioned that the British Engineering Standards Association has laid down standard specifications for automobile steels. The hardness of heat-treated spring-steels varies from 360 to 420 on the Brinell hardness scale.

A typical semi-elliptic spring for a 14 h.p. (R.A.C.) car of about 1 ton total loaded weight would have a span of 30 to 40 ins., camber of 3 to 4 ins., blade width of 2 to $2\frac{1}{2}$ ins., blade thickness of $\frac{3}{16}$ ins. There would be about 9 to 11 blades, each with an overlap of $1\frac{1}{2}$ ins. The shackle pins would be $\frac{5}{8}$ in. to $\frac{7}{8}$ in. in diameter. The typical quarter-elliptic spring used on light cars has a length of 20 to 26 ins., width of $1\frac{3}{4}$ to $2\frac{1}{4}$ ins., blade thickness of $\frac{5}{32}$ to $\frac{3}{16}$ in. and it contains from 7 to 9 blades.

Types of Laminated Springs.—In the past there have been numerous examples of laminated spring arrangements, each with its particular application. Some of these systems have since fallen into disuse, but one or two nevertheless, are of sufficient academic interest to warrant their mention in this section. The common simple types, some of which are illustrated in Figs. 22 and 23 include (1) The Quarter Elliptic, or Single Cantilever; (2) The Semi-Elliptic; (3) The Three-Quarter Elliptic; (4) The Full Cantilever; (5) The Transverse Semi-Elliptic, and (6) The Platform Type.

The quarter elliptic system was previously much used on English light cars for both front and rear springing. It is usual to bolt the front axle rigidly to the lower two leaves, but in the case of the rear axle to provide either a bearing around the axle casing to the outside of which the spring ends are securely attached, or to allow the ends to slide relatively to the axle casing in suitable guides, with stops; this enables the propeller shaft and springs to be of different lengths without either tending to bind when the springs

deflect. One big advantage of this type of spring is the absence of shackles and pins, the lubrication and wear of which are one of the drawbacks of motor-cars. About 10 per cent. of modern cars use quarter-elliptics.

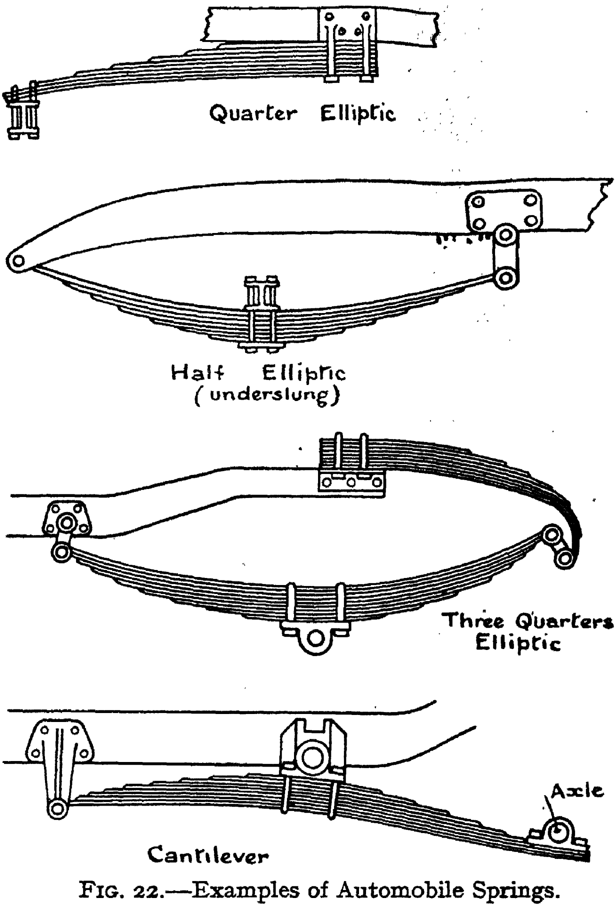


FIG. 22.—Examples of Automobile Springs.

The semi-elliptic system is by far the most popular, at present, no less than 80 per cent. of modern cars employ this type of suspension. We have dealt with the use of the semi-elliptic for the front springs; it now remains to consider the use of these for the rear suspension. When the drive of the rear wheels is taken by the springs to the frame, the Hotchkiss system of hinging the ends is used.

In this case (see Fig 3(*b*)) the front end of the flat type semi-elliptic spring is hinged directly on the frame bearing-bracket; the rear end is provided with a shackle and two pin joints, to allow for the longitudinal motion of the spring. In cases where proper torque tubes or members are provided to take the driving thrust, the semi-elliptic

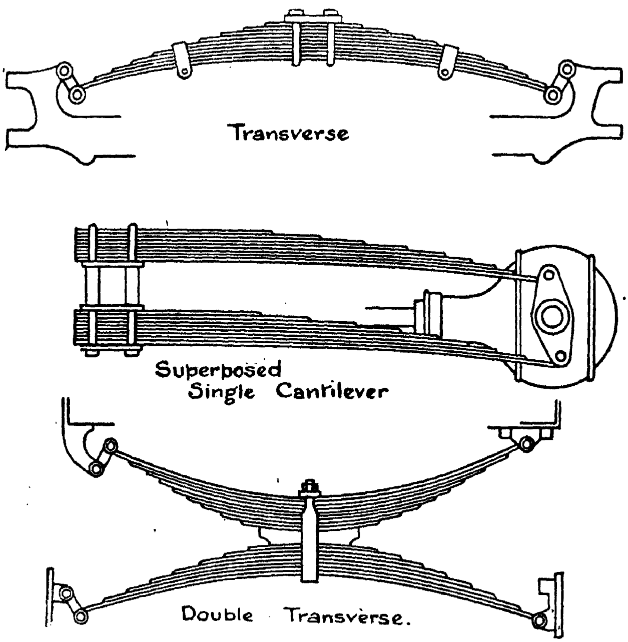


FIG. 23.—Further Examples of Automobile Springs

springs are shackled at *each* end. *The lubrication of the shackle pins* is an *important item*, owing to the heavy loads and rapid rocking movements thereon. Although the grease-gun method of forcing grease through the centres of these pins is most convenient, the regular oiling of the surfaces is preferable when it can be done. The surfaces of the bushes should be provided with oil grooves to distribute the oil.]

In the earlier days of motor-cars the double elliptic system, sometimes seen upon light horse-carts, was common; the *three-quarter elliptic* is intermediate between this and the semi-elliptic.

The three-quarter elliptic is very rarely used to-day for the rear springing, except on one or two earlier makes of taxicabs still on the road. It gives good resilience, but occupies more room than the other systems. It will be noted that the body and frame hang, or depend from, the upper spring, on the shackle member; the latter is thus in tension.

The *full-cantilever system,* illustrated in Fig. 22 has been popular for the springing of certain medium and larger cars. It consists, virtually, of an inverted semi-elliptic, provided with a central trunnion or bearing, the back axle being secured to the rear end. The advantage of this type lies in *the*

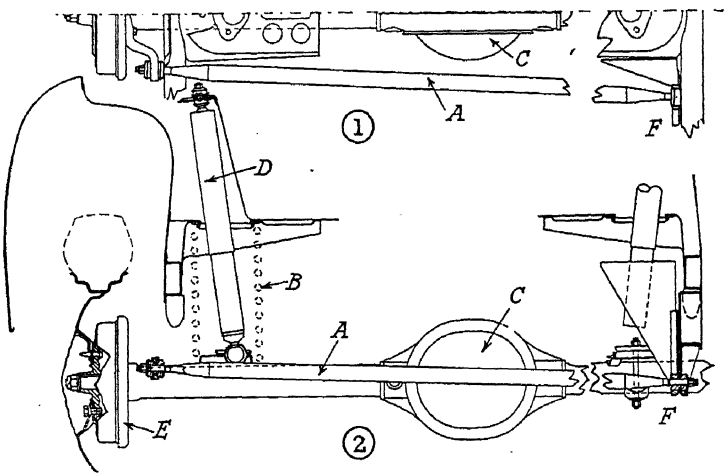


FIG. 24.—The Nash Rear Helical Coil Suspension System. (1) Plan View. (2) Rear End View.

appreciable reduction in the unsprung weight, for most of the weight of the spring is carried on the frame; similarly, also, with the quarter elliptic. Where the trunnions occur on the frame, it is usual to place a stiff tube or cross-member in the chassis frame, to stiffen the latter. The front ends of the springs are shackled as a rule, but in some cases one end is allowed to slide between suitable rollers, or guides. In the Wolseley cantilever system there was a rocking shaft right across the frame connecting the central parts of the two springs, the object being to transmit to both springs equally any road shocks received by either;

this assists greatly in damping the oscillations or movements out.

Transverse springs are sometimes used for lighter makes of car, the Ford and Austin (8 h.p.) being examples. There the front springs are placed transversely to the side-members of the frame; the latter is attached to the centre of the semi-elliptic shape spring and the axle is connected through shackles to the other ends; the shackles allow the flattening of the spring to occur freely under load. In the case of the Ford car, transverse springs are placed both fore and aft; for heavier or faster cars, this two-point attachment of the frame is apt to cause rolling of the body on corners and on irregular roads. It is important, when transverse springs are used to fit radius rods to the two ends of each axle, and to a hinge on the main frame so as to preserve, or locate, the lateral position of the axle under all circumstances. In the Austin (8 h.p.) car the transverse spring is used only for the front axle springing.

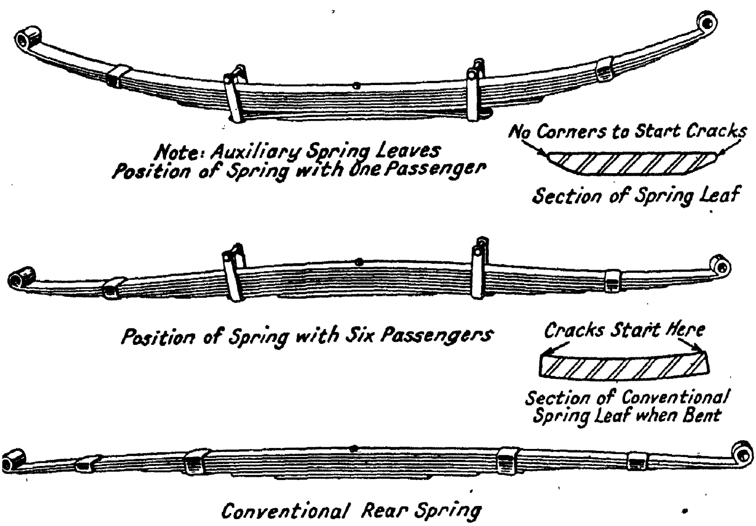


FIG. 25A.—Illustrating Leaf Spring Improvements.

Leaf Spring Improvements. Passenger car springs are now being made with the "helper" spring method of automatic load adjustment as shown in the two upper illustrations in Fig. 25A. When the car is lightly loaded

the spring shape is similar to that given in the top illustration and when fully loaded to that shown in the centre illustration. The lower view shows the conventional shape previously employed.

Another improvement is the use of bevelled sections for the spring blades instead of flat ones as shown in lower right hand diagram; the former shape has been found to be immune from the tendency to start cracks at the corners.

Helical Coil Spring Rear Suspension. Helical springs have been used in certain American cars for the rear suspension, a typical arrangement employed on the American Nash cars, being illustrated in Fig. 24. The ordinary pattern rigid rear axle is employed and the helical

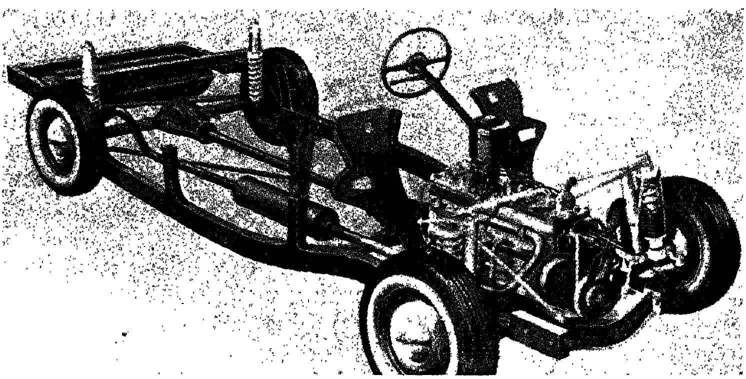


FIG. 25.—The Nash Front and Rear Helical Coil Springing System.

springs B, which are of relatively large diameter are located between the ends of the axle casing C and the side frame members of the chassis. In order to relieve the springs of any driving or braking stresses, so that they can act only as springing members, in the case of a chassis having a torque tube, braced to the axle by stays, this member forms the longitudinal radius arm and lateral stability is established by a transverse radius rod A pivotally connecting a bracket on the frame to a bracket F on the axle casing. Each end of this rod is mounted on resilient bushes. Telescopic hydraulic shock absorbers D between

the axle spring seats and the frame members are also resiliently mounted. Two longitudinally arranged radius arms are provided when the chassis has an open propeller-shaft or a torque tube furnished with a sliding joint. Although shown applied to a rear axle the arrangement can also be employed for front axles with steerable wheels.

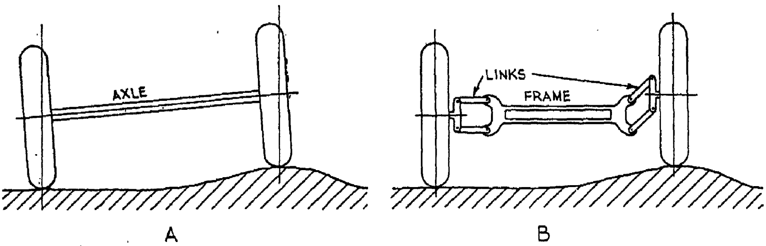


FIG. 26.—(A) Rigid Front Axle Springing. Axle Tilted when One Wheel encounters Road Bump. (B) Independent Springing. Frame remains horizontal and Wheels Vertical.

Independent Wheel Suspension. [The method of employing a rigid front axle with semi-elliptic springs near its ends whilst being an economical arrangement from the viewpoint of car manufacture, is open to certain objections, namely: (1) Any springing movement of one wheel is communicated through the axle to the other, the body of the car being tilted, accordingly; (2) The unsprung weight is appreciable, and (3) The system is liable to result in wheel wobble. In order to overcome these drawbacks and, at the same time, to provide a greater degree of vertical springing movement the method of springing each wheel independently has been adopted in several commercial car models.]

The method is not new, however, since it was used in pre-War days and subsequently on the rather later Morgan, Lancia (Lambda), Sizaire and Beck cars. In recent years, however, better methods of achieving the same result have been developed.

There are now several different methods of springing each front wheel independently, some typical ones being shown diagrammatically in Fig. 27. It should be mentioned that laminated and also coil springs are employed in these systems; in addition, shock absorbers of the friction and

hydraulic types are used between the moving wheel members and the fixed chassis frame.

In connection with the general design of independent front wheel suspensions it is important to arrange the wheel restraining mechanism so that this will permit the wheel to move vertically up and down; otherwise, if

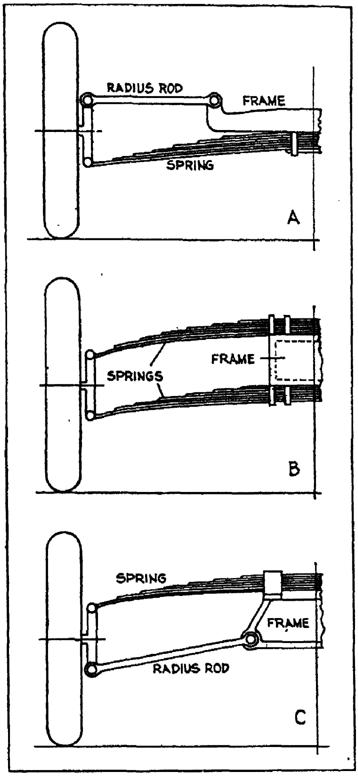


FIG. 27.—(A) Lower Transverse Spring.
(B) Double Transverse Springs.
(C) Upper Transverse Spring.

there is any lateral or tilting action on the wheel, tyre drag, or scruffing action will result, with detrimental effects in regard to tyre wear and possible skidding tendency.

In the case of most independent wheel springing systems the unsprung weights are appreciably lower than for the

fixed axle ones, so that the springing action can be made more efficient.

In regard to the action of the front wheel brakes, the brake reaction has to be taken by the independent spring members, so that these must be adequate in strength and

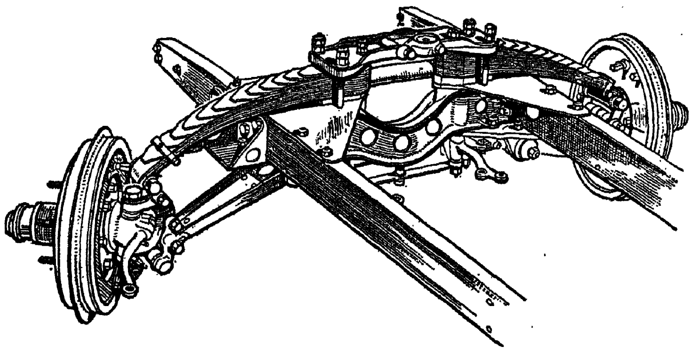


FIG. 28.—Transverse Leaf Spring Method.

rigidity for this purpose. The guide or radius arms which serve to maintain the wheel in the vertical position under springing action, are usually duplicated or made

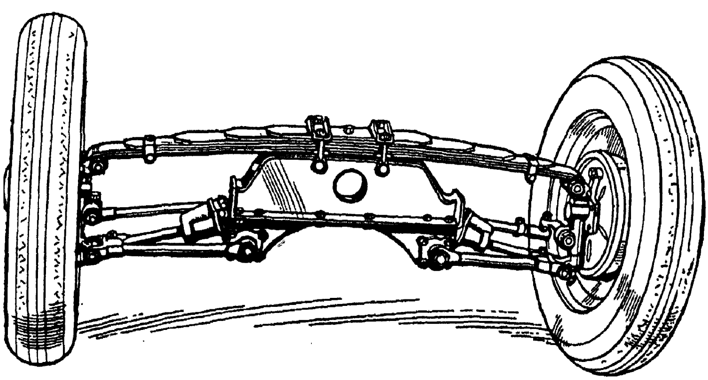


FIG. 29.—Fraser Nash B.M.W. Springing System.

triangular for this purpose; the so-called "wish-bone" radius arm is a typical example of this.

In some cases one of the two radius arms is replaced with a transverse leaf spring, a good instance being the

springing system shown in Fig. 28.* Here, a kind of inverted semi-elliptic spring is clamped securely over its central portion to the chassis frame, the ends being hinged to the upper parts of the stub axles. The lower links form parallelograms, one on either side thus ensuring correct vertical movement of the wheels.

In the example shown in Fig. 29, the laminated transverse spring is also mounted at the top, the forked designed radius arms being below.

Another method, which is employed for the front driven B.S.A. four-wheeler cars, uses laminated springs in pairs, for both the upper and lower stub-axle members. There are thus two laminated springs—in the same horizontal plane—above, and two below. In addition there is a relatively large friction type shock absorber with its outer end hinged to the top of the stub axle unit and its inner friction jointed one hinged to a frame mounted bracket; it therefore acts as a kind of radius arm as well as a shock absorber.

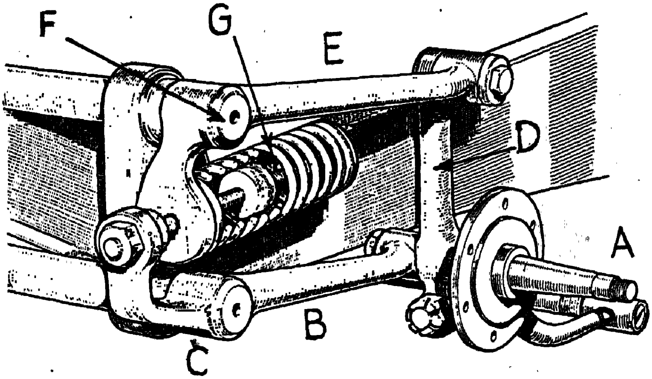


FIG. 30.—Gordon Armstrong Independent Springing Method.

The Gordon Armstrong System. In this method, instead of having radius arms moving in a lateral plane—as in the examples previously described—these arms move in a longitudinal sense, thus providing a compact arrangement at the front of the car. Referring to Fig. 30† the wheel axle A is carried at the end of the lower radius member B, which can rock about a fixed hinge C, carrying on its

* *The Autocar.* † Reproduced by permission of *The Motor.*

other side a bent arm providing an anchoring for a rod which is connected through the centre of the spring G to its outer right hand side. The other members are a connecting-rod D hinged at its lower end to B and another rocking member E having a fixed position bearing at F and an extension beyond against which the other end of the coil spring abuts.

When the wheel is forced upwards under the influence

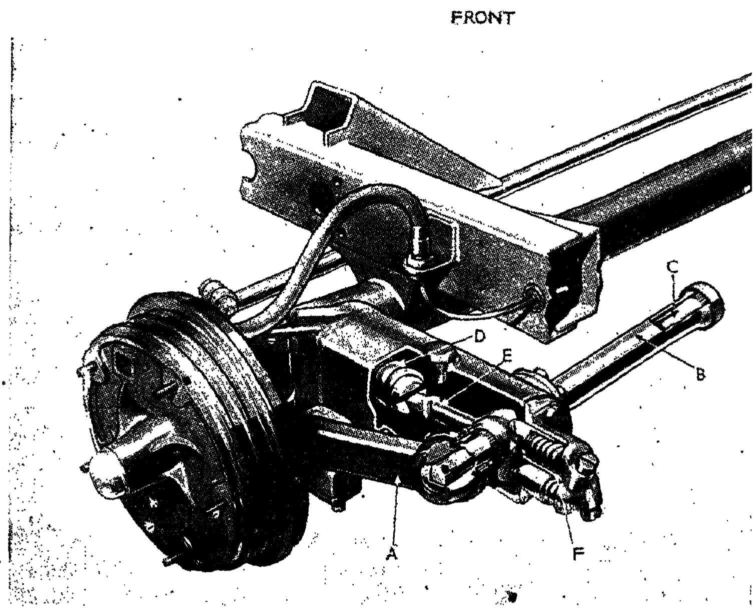


FIG. 31.—Vauxhall Independent Front Wheel System, with Torsion Bar Springing.

of road inequality, the link D moves upwards, causing the left-hand end of the member E to rock so as to compress the spring whilst the left hand end of member B, through its tension rod also compresses the spring, thus affording the necessary resistance to the vertical movement of the wheel.

The Vauxhall front springing system, originally fitted to the 12 and 14 h.p. cars operated upon a somewhat similar principle, the wheels being carried on the front

ends of short arms hinged at the rear to fixed chassis members enclosing powerful coil compression springs which were acted on by cam-like projections on the other sides of the wheel arms, inside the spring casing. Hydraulic shock absorbers were also employed; these were located inside the coil springs. Below, and parallel with the wheel arm, in each case was another radius rod to take the brake reaction effect. In the more recent Vauxhall front springing system the torsion bar method is employed, in conjunction with a variable rate of springing control and spring shock absorbers.

Referring to Fig. 31 showing a cut-away view of the system, it will be seen that the wheel (the hub only being shown) is fixed to the wheel carrier arm *A*. When this arm rises or falls with the movements of the wheel it twists the torsion bar and tube (*C* and *B*); this part of the suspension is designed to take the greatest road shocks. On fairly good roads the coil spring *D* bearing

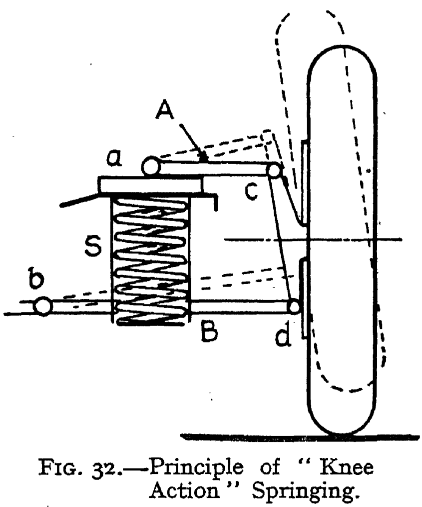


FIG. 32.—Principle of "Knee Action" Springing.

upon the toggle *E* is arranged to assist the wheel arm in twisting the torque bar and tube; the full effect of this spring is only felt, however, when the wheel movements are relatively slight. When bad surfaces are encountered the toggle moves farther away from the dead centre and the force exerted by the spring becomes relatively less,

thus giving, in conjunction with the torque tube, a gradual stiffening suspension with increasing movement of the wheel arm. This arrangement is claimed to provide

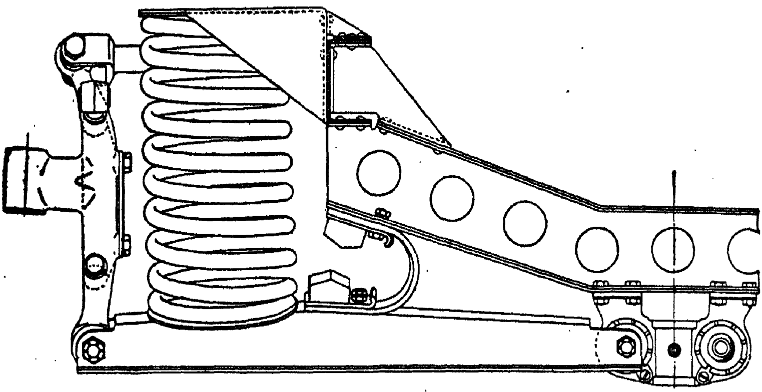


FIG. 33.—Andre-Girling Springing Method Employed on Daimler and Lanchester cars.

stability of suspension when the car is swerving or cornering, and thus to prevent the rolling action which is characteristic of certain independent front wheel suspension systems.

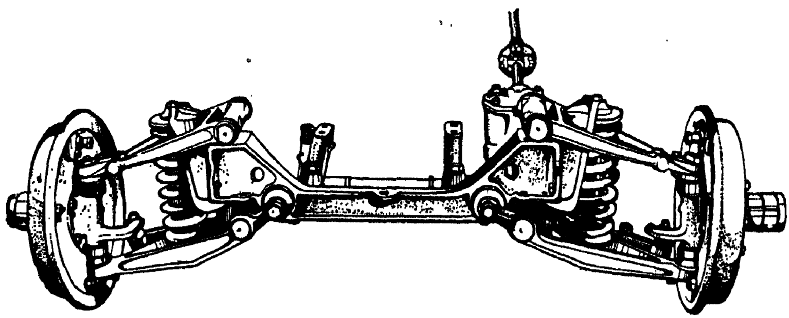


FIG. 34.—Mercédès Benz Springing System.

'Knee-Action' Springing. The so-called 'knee-action' springing method used on the Mercédès Benz, Buick and other American cars employs a pair of radius arms *A* and *B* (Fig. 32) with hinge centres on the chassis frame at *a* and *b* and other hinges at *c* and *d* on the stub axle. The springing effect is obtained by the compression of a large diameter coil spring *S*, when the wheel system moves upwards. As

the upper radius arm is shorter than the lower one the wheel tilts inwardly at the top, when it strikes a road projection —as indicated by the dotted lines. As the lower arm moves the lower part of the wheel inwards, this action combined with the tilting effect maintains the wheel track practically constant, so that 'scuffing' is avoided.

Fig. 34 illustrates a typical springing system based upon this principle; it will be observed that the radius arms are of the previously mentioned 'wish-bone' pattern. The system also includes double compression spring shock-absorbers.

The Rolls Royce independent front wheel springing system shown in Fig. 35 also uses the same general principle,

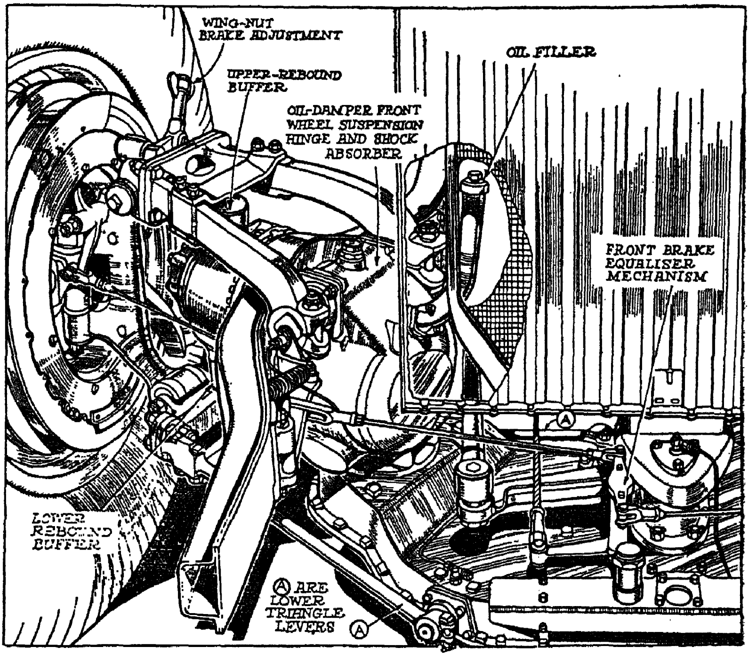


FIG. 35.—Rolls Royce Independent Front Wheel Arrangement.

but with the spring arranged laterally in an oil-tight casing, with hydraulic shock-absorber. Special rebound buffers are fitted at the upper and lower sides. This illustration shows also the built-up box-section chassis rame (on the lower left-hand side).

Semi-Elliptic Springs for Independent Suspension.—A method of springing each front wheel independently,

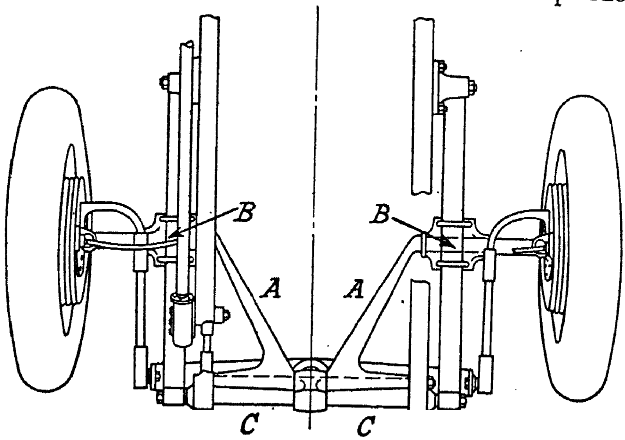


FIG. 36.—The Nicholas Front Suspension System.

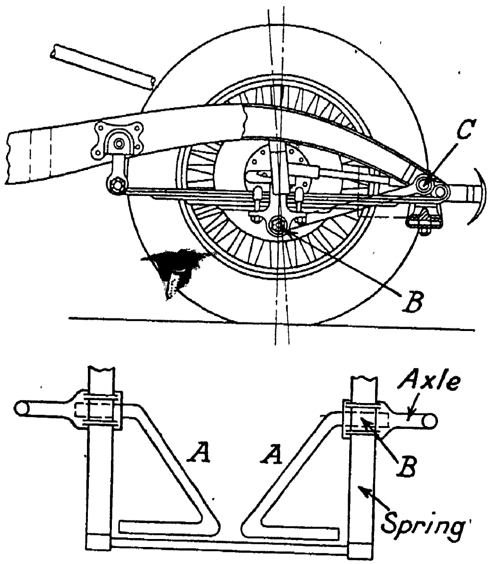


FIG. 37. (Above.) Side View of Nicholas Suspension System.
(Below.) Shape of Hinged Link A.

known as the Nicholas, employs the usual fore-and-aft disposition of the semi-elliptic leaf type of spring. Each

wheel is mounted on the end of a link of special shape, resembling three sides of a rhomboid, as shown at A in Figs. 36 and 37*. The link is attached to the lower part of the centre portion of the leaf spring at B and its other end is made as a shaft which can rock in a bearing C formed in a rigid cross member of the frame, at the front. Thus, as the wheel moves up and down under springing action the link member hinges about the bearing C and is constrained by means of the spring.

This system has the advantages of simplicity, great rigidity, increased stability when cornering and similar long wearing properties of the bearing members to those of the orthodox leaf springing system. It gives, however, a relatively heavy unsprung part of the springing system.

Transverse Independent Springing. A method of converting a chassis having the ordinary transverse front leaf spring (such as the Austin or Ford cars) into

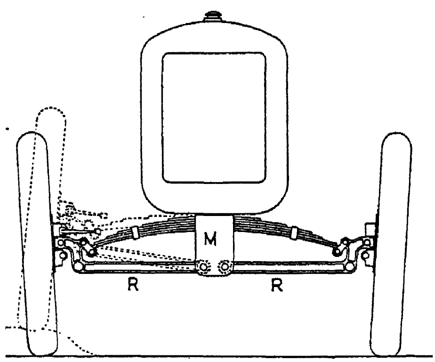


FIG. 38.—Transverse Springing Conversion.

independent wheel springing is shown in Fig. 38. In this case the rigid front axle is replaced with the fixed central bracket *M*, rigidly secured to the chassis frame, and two radius rod members *R*, hinged to *M* and connected, as shown to the stub axle and leaf spring ends by means of the usual short shackles. When one wheel strikes a road obstacle it takes up the position shown by the dotted lines on the left, the other wheel remaining unaffected;

* *The Automobile Engineer.*

this is a definite improvement upon the fixed axle springing method normally employed, although it results in a certain 'scuffing' action and wheel tilt.

Notes on Independent Springing Systems. Generally these systems are more complicated in construction than the orthodox fixed axle ones and, as they usually involve several additional bearings, the maintenance attention is greater; moreover, when wear occurs the effects upon the springing and steering are often more pronounced.

With well-designed systems, however, the drawbacks of 'wheel wobble' and 'steering tramp' are avoided entirely.

Unless, however, the springing and steering systems are correctly designed there will always be some interaction between the two so that when cornering, for example, the car may tend to skid; or when the wheels are traversing bad road inequalities the steering may be adversely affected.

Another possible disadvantage with some front wheel systems is the gyroscopic effect due to the plane of the wheel tilting from its normal vertical position; this results in a turning action in another plane which results in a steering 'tug' being experienced. As both the front wheels are involved a series of transverse tugging actions on the steering will occur, giving a somewhat similar action to that experienced with 'wheel wobble' on fixed front axle cars.

On the other hand, by maintaining the wheels in the approximately vertical plane, this effect can be avoided.

A well-designed independent wheel system enables springs of greater resilience to be used, giving much better springing action than with most rigid axle cars.

Independent Rear Wheel Springing. The advantages of independent front wheel suspension over the orthodox rigid front axle method are so well established that attention has now been given to independent rear wheel suspension, since it is recognised that the ideal system is for each wheel of a vehicle to be sprung separately.

The application to the rear wheels is rather more complicated than with the front ones on account of the power drive to the former. This problem has been solved in

the case of front drive cars, however, so that the rear wheels—which do not have to be steered, as with the front members—should not prove difficult.

From the economical point of view, independent wheel suspension is more costly both in the initial expense and

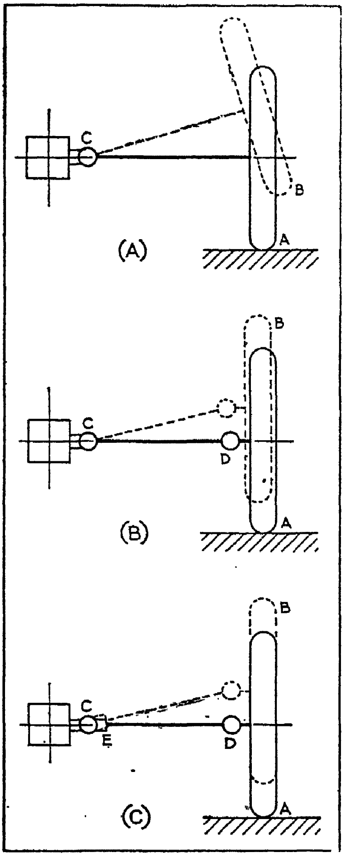


FIG. 39.—Rear Wheel Independent Springing Methods.

in maintenance—on account of the more complicated mechanism and increased number of wearing surfaces. Considering the fundamental points of rear wheel independent springing there are several alternative methods possible, including entirely separate drives to each rear

wheel and a single central power drive with two take-off shafts to the rear wheels, in the same manner as most front drives. Referring to Fig. 39 (*A*) showing a central back axle and differential unit driven by the usual propeller shaft, this unit is mounted rigidly to the chassis frame. Each rear wheel is driven by a cardan shaft with a universal joint at *C*, the axle being attached solidly to the wheel or hub in order to drive it. The full line *A* shows the normal wheel position, whilst the dotted line *B* shows the position taken up under springing action; it will be observed that the wheel track has increased so that outward skidding or 'scuffing' must occur. As an improvement a second universal joint may be introduced, as shown at *D* in Fig. 39 (*B*). In this way the plane of the wheel may be constrained (by suitable spring arrangement) so that it moves in a vertical direction and with reduced lateral movement; there is, however, still a scuffing action.

In order to overcome this difficulty it is necessary not only to apply universal joints at each end of the

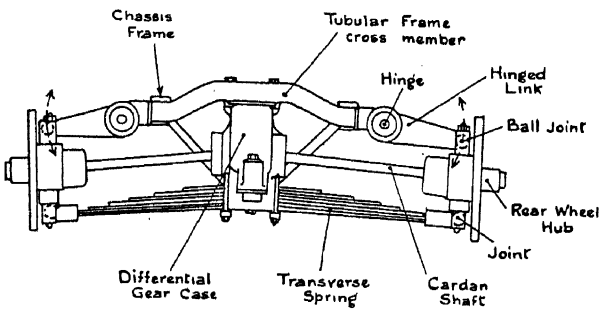


FIG 40.—Separate Rear Wheel Springing System.

cardan shaft but also to introduce a sliding or 'plunging' coupling at one end, in order to allow for the lateral action or to allow the springing system to maintain the wheel track constant. The arrangement shown in Fig. 39 (*C*) shows the principle of this method, the plunging coupling being indicated at *E*; in this example the wheel is constrained, by suitable radius rods parallel to the longitudinal axis of the car, to move vertically, there being no lateral pull due to the drive.

One method of obtaining approximately vertical wheel movement is by means of a transverse leaf spring with

hinged radius arms above as shown in Fig. 40. The radius arms may conveniently be combined with friction or hydraulic shock absorbers. The cardan shafts, in this case have universal couplings at each end and a plunging coupling at one end.

Another rear drive system but without independent springing shown diagrammatically in Fig. 41 follows the

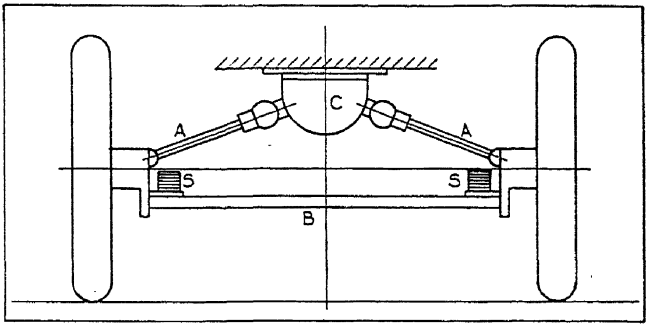


FIG. 41.—Another Rear Wheel Drive Method.

earlier De Dion method of mounting the differential casing *C* rigidly on to the rear cross frame members and using cardan shafts *A*, each with a pair of universal couplings and a plunging coupling to drive the rear wheels. The latter have fixed members for locating the brake mechanism, the power drive bearings and spring pads, these fixed members being connected laterally by means of a light tubular or channel member *B*; this ensures that the rear wheels will move together as in current practice. As the connecting member can be made considerably lighter than the usual back axle casing and its components, the unsprung weight can be greatly reduced by this method.

Springs in General.—In order to provide as low a frame (and body) position as possible the axles of the car are often attached to the upper, instead of to the lower sides of the semi-elliptic springs; thus in Fig. 22 the axle is shown fixed above instead of below; this type is known as the *Underslung* one.

In the *Platform* method of springing used on one or two American cars, including the earlier model Cadillac, a pair of semi-elliptics is employed for the rear springing, but

their rear ends, instead of being connected to the frame, are attached through universally jointed shackles to the ends of a transverse spring, the centre of which is attached to the rear frame cross-member. (Fig. 23.)

The Citroen car previously employed duplicate quarter-elliptic springs at the rear, the rear axle being positioned at the centres of the shackle plates hinged to the rear ends of the springs; the latter took the torque reaction, the upper and lower springs being in tension and compression respectively, due to the torque.

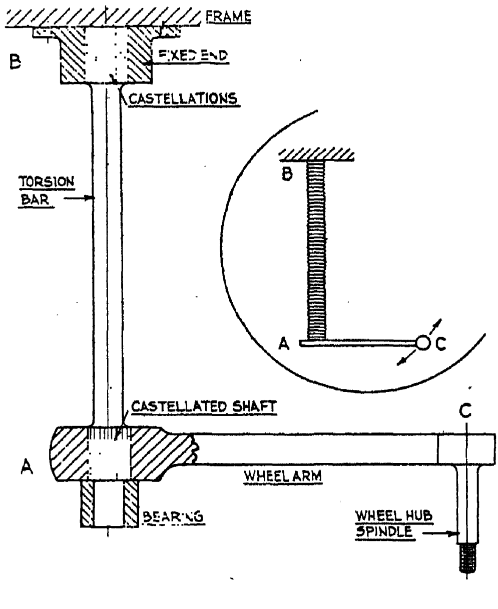


FIG. 42.—Principle of Torsion Bar Springing.

Torsion Bar Springs. Torsion bars, for springing purposes have been used in conjunction with leaf springs, or alone on several makes of racing and, more recently on certain British and Continental production model motor cars.

The principle of torsion bar springing is illustrated in Fig. 42. Referring to the right-hand inset diagram a spiral spring is shown fixed at one end *B* and having an arm *AC* fixed at *A* to the spring. It will be observed that

when any tangential force is exerted at *C* in a plane at right-angles to the axis of the spring it will tend either to twist or untwist the spring ; in the former case the torsion of the spring opposes the force applied at *C*.

Referring next to the left-hand diagram, in place of the spring is shown a solid bar of steel having castellated or keyways at the end *B*, secured rigidly to a frame bracket ; the other end *A* is secured rigidly to the end of the wheel arm *AC*. A bearing is provided at *A* to prevent any bending action on the torsion shaft. The end *C* of the arm carries the wheel spindle and hub.

When the road wheel on this spindle receives a shock the arm *CA* tends to rotate so that *C* moves upwards thus tending to twist the torsion bar *AB*. The latter is made of a strong alloy steel and is so designed that it can twist through a small angle (equivalent to a few inches vertical movement of the wheel) whilst the stresses in the steel are well within the elastic limits—or more accurately the impact fatigue stress limits. The bar thus acts as a torsion spring in resisting the springing loads.

The weight of the car itself causes an initial twist, just as the car's weight gives ordinary leaf springs an initial set ; with most torsion springing systems an adjustment is provided for varying this initial torque bar setting in order to raise or lower the wheel positions relatively to the chassis frame.

The advantages of the torsion bar springing method are its comparative lightness and the small space which its components occupy. It has no wearing parts (except the outer bearing), since the system is a rigid one and it can be arranged either transversely as in the Vauxhall and Auto-Union cars or parallel with the frame.

A further advantage is that, owing to its lightness, the unsprung weight is reduced to a minimum.

It should be mentioned that in some instances the torsion bars are replaced by torsion tubes ; for the same torque values the latter can be made much lighter than solid torque members.

The disadvantages of the torsion bar method as compared with leaf springing systems are that there is no frictional damping, so that some external damping device or special design of shock absorber is necessary in order to give this

desirable spring damping action which, incidentally, must operate effectively for very small wheel movements.

Another drawback of the torsion bar system is that it does not take the braking or driving torque so that additional arms or linkages become essential for this purpose.

The Vauxhall torsion springing system, shown in Fig. 31 employs a kind of double torque tube, one tubular member —carrying at its lower end the front wheel—being taken inside another torque tube and attached to it at its upper

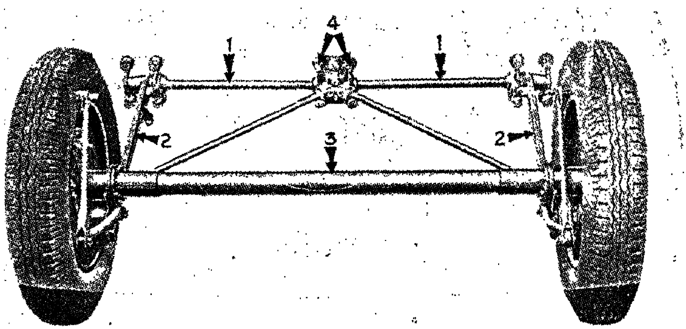


FIG. 43.—Citroen Rear Suspension System.

end. The outer tube is anchored at its other end to the chassis frame.

The Citroen torsion springing system shown in Fig. 43 for the rear wheel springing (the front wheels* being power-driven in this car) employs torsion bars arranged somewhat on the lines of Fig. 42. The rear axle casing is continuous, since the rear wheels are trailing ones only. The axle 3 is connected by longitudinal arms 2 to the torsion bars 1, suitable bearings being provided at the outer end of these bars to obviate bending. The inner ends of the torsion bars are rigidly located to a central member 4 attached to the rear transverse frame member, an adjustment being provided to each anchoring so that by means of a screw it is possible to vary the car height relatively

* See also Fig. 309 on page 364.

to the road. Friction type shock absorbers are fitted at the front, and hydraulic ones at the rear of the chassis.

The front wheels are sprung by torsion bars, but these are arranged with their axes parallel to the longitudinal axis of the chassis; the torsion bars lie close to the frame side members, being anchored at their rear ends to the frame brackets.

In connection with the location and fixing of torsion bars to their anchorings or wheel arms, this is generally

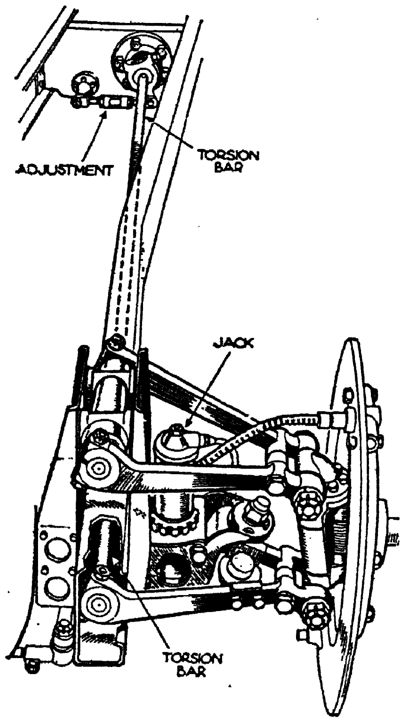


FIG. 44.—Lagonda Torsion Springing System.

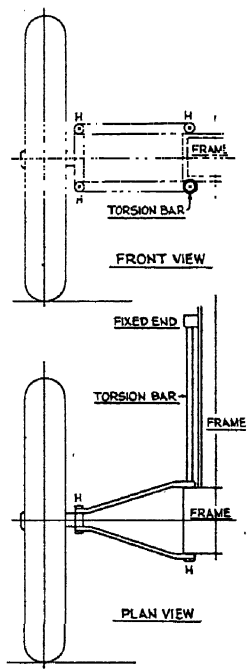


FIG. 45.—M.G. Racing Car Springing System.

done by splining the enlarged ends of the torsion bars; accurate machining and fitting is essential as, otherwise, any slight backlash will cause much free movement of the wheel as well as noise.

The longitudinal torsion bar springing method used on one of the successful M.G. racing cars is shown in Fig. 45.

In this case both front and rear wheels were sprung independently on wishbone arms, the upper ones being hinged and the lower inner members attached to torsion bars. The ordinary hinged joints are indicated by the letters *H*. The general arrangement of this car, in Fig. 17, also shows the front and rear torsion bar springing systems.

The independent front springing used on the Lagonda Vee-12 engine car is similar in principle to that illustrated in Fig. 44. It employs a long torsion bar for the two lower radius arms and a plain bearing for the two upper arms. An adjustment for wheel alignment purposes is provided at the anchorage of the torsion bar. The illustration shows also the hydraulic jack, which is provided at each wheel, for the purpose of raising the wheels clear of the ground.

Rubber-Air Springing.—A more recent front and rear springing method for which is claimed a 'softer' springing action comparable with that of the best independent wheel systems, is that devised by The Firestone Tyre and Rubber Company, America. It comprises an inflated rubber spring consisting of a rubberised fabric bellows filled with air at a pressure corresponding to the load to be carried. The bellows operates automatically in conjunction with an air reservoir, by means of a pendulum shock absorber valve; the reservoir and bellows are connected by means of metal tubes. The device weighs about 2 lbs. per wheel, compared with anything from 15 to 50 lbs. for steel springs.

It is stated that the results of road tests have shown a more comfortable springing action to result; less tendency to roll when cornering and quieter operation; the maintenance attention of this suspension device is much smaller than for ordinary mechanical systems.

Phased Suspension Systems. In connection with the design of semi-elliptic springing systems for both the front and rear of the chassis, it is possible by making the vibration frequencies of the front and rear springs different by estimated amounts, to improve the riding qualities of the car, appreciably. If, for example, the front and rear springs had the same frequency the springing would be of a very 'bouncy' nature, but by making the frequencies

different they may be arranged to cancel out spring resonance or bounce effects to a large extent.

Fig. 47 illustrates the front spring *a* and rear spring *b* of

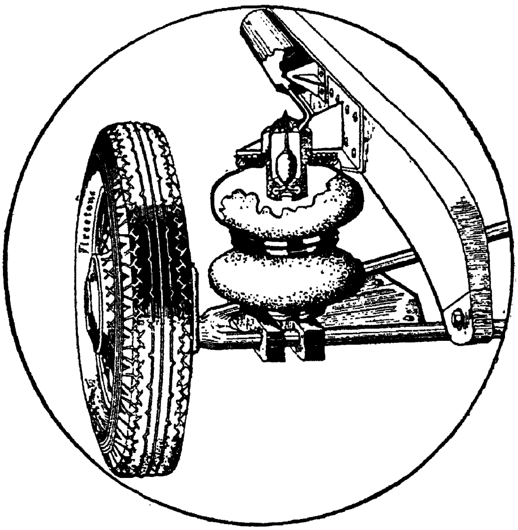


FIG. 46.—The Firestone Rubber-Air Suspension System.

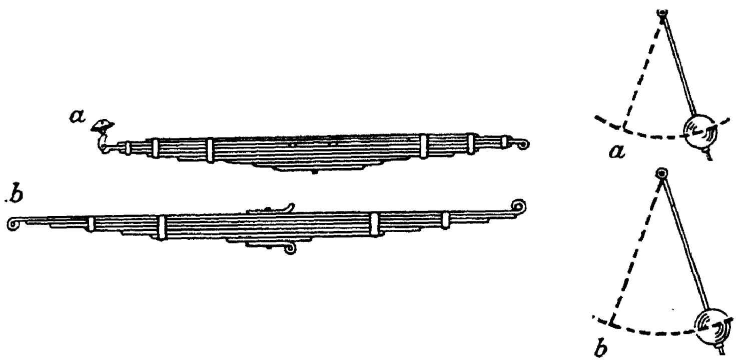


FIG. 47.—Showing Principle of Wolseley Phased Suspension System.

the Wolseley suspension system which have relatively high and low natural vibration frequencies, corresponding to those of the two pendulums shown on the right. When the

car strikes a bump in the road the front and rear springs tend to move at different frequencies which give a cancelling effect at normal car speeds so that the car experiences a stabilizing action.

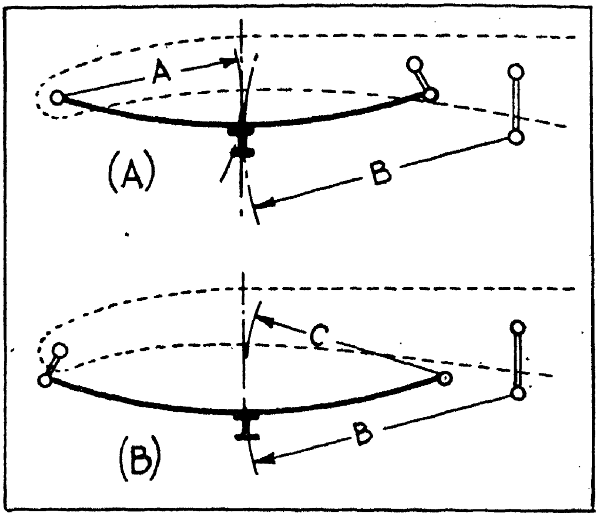


FIG. 48.

Spring Shackle Positions.—With the conventional position of the shackle at the rear end, in the case of front semi-

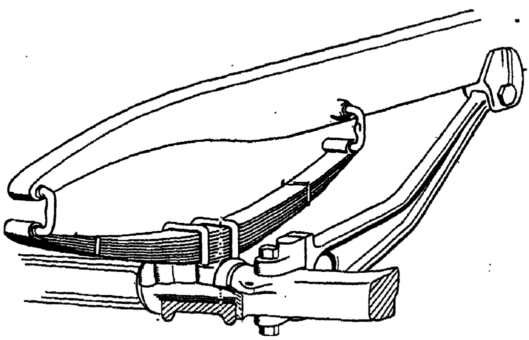


FIG. 49.—Springing Method Employing Radius Rod, both ends of Spring being Shackled.

elliptic springs, the effect of spring deflection is to cause the front axle to move upwards about a different (and

shorter) radius than the steering drag link. Thus, in Fig. 48 (A) the axle will tend to move in an arc of radius *A*, whilst the drag link *B* can only move in one of radius *B*; the result is to cause a steering effect on the wheels giving rise to steering wobble.

If, however, the shackle is arranged at the front end (Fig. 48 (B) then the front axle will move in an arc of radius *C*, which can be made to be approximately the same as the arc of radius *B*; in this case there is practically no relative movement between the axle and front drag link attachment so that more stable steering is effected. When this arrangement is employed a radius arm (or duplicated arms) as shown in Fig. 49 may be used.

Extra Load or Helper Springs.—A design of laminated spring that is used, mostly on commercial vehicles, in order to provide for variable loads or progressive springing action is shown in Fig. 50. It will be observed that, in

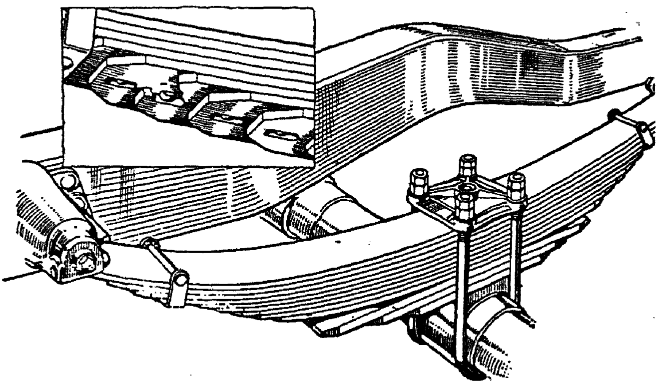


FIG. 50.—Helper Springs, used on Morris Commercial Vehicle.

addition to the usual leaf spring combination, there are, in this example, three spring blades below; these are clamped between the spring and axle.

When the vehicle is only lightly loaded these 'helper' blades do not come into operation, but as the load is increased the main spring deflects and the other blades then take their share of the load and contribute to the road springing action.

Torsion Bar Anti-Roll Device.—In order to prevent rolling of the car when cornering at any appreciable speed, due to the effect of centrifugal action in most normal cases where the centre of gravity is not very low, the springs on the opposite sides of the chassis are connected by a torsion bar device. The effect of this is to couple the two springs so that the one that is normally inactive now contributes towards the springing resistance to rolling or excessive road inequality on the one side. The torsional resistance of the bar itself is also arranged to oppose the rolling tendency.

The principle of the method is illustrated in Fig. 51. In this example the two springs *A* and *B*, which are

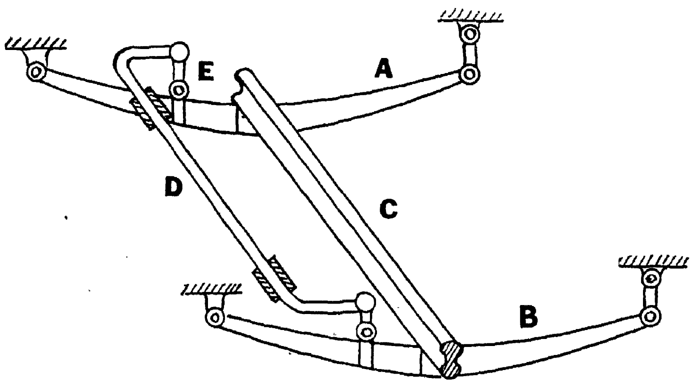


FIG. 51.—Torsion Bar Anti-Roll Device.

attached to the axle *C* are connected by means of short jointed rods *E* to the ends of a torsion bar *D* having cranked ends and fixed position bearings on the chassis frame. When one spring only is deflected, one of the rods *E* tends to apply a torque to the bar *D*, so that it not only experiences a twisting action but transfers part of the load to the other spring by means of the other rod *E*.

Silent Spring Ends.—It is a well-known fact that the ordinary spring shackle-pins and their bearings become noisy after they have been in use for some time, due to wear. In order to overcome this drawback attempts have been made to do away with the shackle-pin bearings and to substitute flexible suspensions of the fabric or rubber types.

The most successful of the latter are the Silentbloc rubber bushes now fitted to a number of different makes of car. In this case there is an inner and an outer metal bush

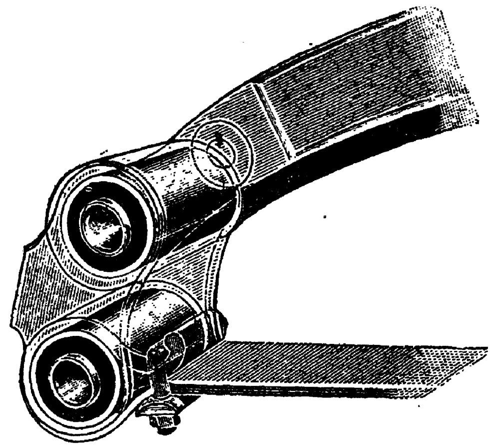


FIG. 52.—Silentbloc Spring Bearings.

separated by rubber in compression. The shackle-pin is a fairly tight fit in the inner bush, so that when the spring flexes the pin and bush rock together and the rubber

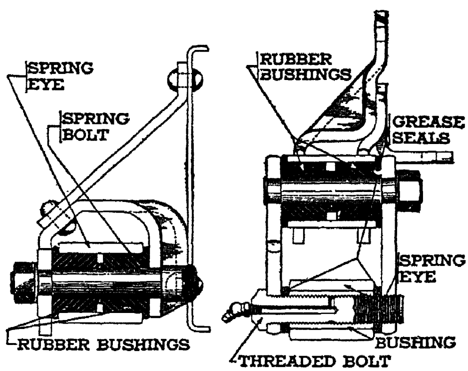


FIG. 53.—Rubber Bushings Used on American Packard Cars.

stretches in a torsional manner. (Fig. 52.) These bushes are silent in operation and require no oiling—indeed, the presence of oil is a disadvantage.

C*

The method of anchoring the spring-end, shown in Fig. 54, utilises a metal bracket which is riveted to one end of the spring-plate. The bracket is surrounded by a moulded rubber block held in a metal casting bolted to the side of the chassis frame. The spring can, therefore, flex in all directions one end of the spring is usually allowed to slide in its metal housing to allow for the extension of the spring under load.

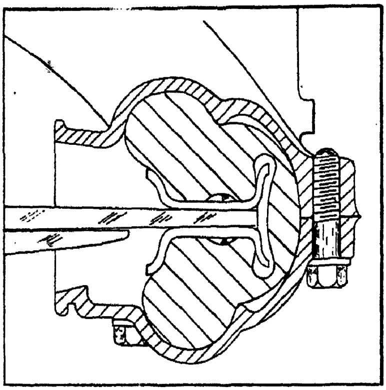


FIG. 54.—Solid Rubber Mounting for Spring Ends.

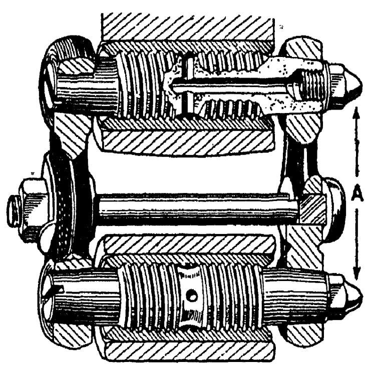


FIG. 55.—Austin Spring Shackle with Threaded Pins.

Shackle Pin Design.—In order to provide the maximum amount of wearing surface between the shackle pin and its fixed bushes in the spring-end and shackle bracket member it is now usual to make the outer surface of the pin of screw-thread form, with a fine-pitch deep section form of thread; in some instances the knuckle form of thread is employed. The inside of the bush is screwed in a corresponding manner so that the pin can rock within the bush. The amount of side movement is so slight that it does not affect the alignment of the springs or wheels.

Fig. 55 illustrates the export type of Austin spring shackle. The method of lubrication of the bearing surfaces is by means of the grease nipples shown at A; the pins are drilled for the passage of the lubricant as shown in the upper pin.

Weight Adjustment.—In the case of light cars, the springs are usually designed so as to function correctly when the car is fully loaded, i.e., with three or four persons.

When used by the driver only, the springing is usually too harsh, and the car does not hold the road so well—it tends to bounce a little. In larger and heavier cars, where the passenger's weight is a smaller proportion of the whole weight this effect is not felt to any extent. Many cars—generally the more expensive ones—are now fitted with adjustable devices for altering the stiffness of the rear springs to suit the load; typical examples include the methods which have been used on the Armstrong-Siddeley, Hotchkiss and Lavoie cars; in the latter cases the adjustment is automatic, the effect of extra weight causing the springs to shorten in effective span, and thus to become stiffer. Another example is shown in Fig. 25A.

Earlier Type Shock Absorbers.—This name is applied somewhat loosely to auxiliary springing devices fitted to the main springs to absorb the smaller shocks without the latter springs coming into action, and is also used to denote auxiliary dampers, fitted for increasing the friction or damping action of the main springs. The simplest example of the former device is the single or double compression springs fitted between the shackle plate and the spring-eye pin bearings; an example is shown in Fig. 56 (1). Damping devices include those working either by solid friction as in the Hartford and Gabriel types, or hydraulically as in the Houdaille oil damper. It is usually agreed that *the function of a 'shock absorber'* is to allow the axle to move freely upwards, but to damp its return action so as to rapidly reduce the main spring oscillations. The Gabriel 'snubber,' Fig. 56 (3) which was once popular consisted of a strip of strong fabric or balata belting attached at one end to the axle, and at the other through a spiral winding to the frame; two semi-cylindrical blocks are fitted inside the spiral and pressed outwards by an internal spring. When the axle is deflected upwards by a road bump, the compression spring takes up the slack in the belt, but when the axle descends the increased friction between the coils increases the damping effect, by offering a resistance to the axle's downward movement.

In the Houdaille shock absorber which was the forerunner of several late hydraulic types, Fig. 56 (4) there was a pair of arms hinged to one another at one end, and attached to the frame and axle (or the main spring) at the

other respective ends. There were friction discs arranged in the hinge joint, to offer a damping action to the movements

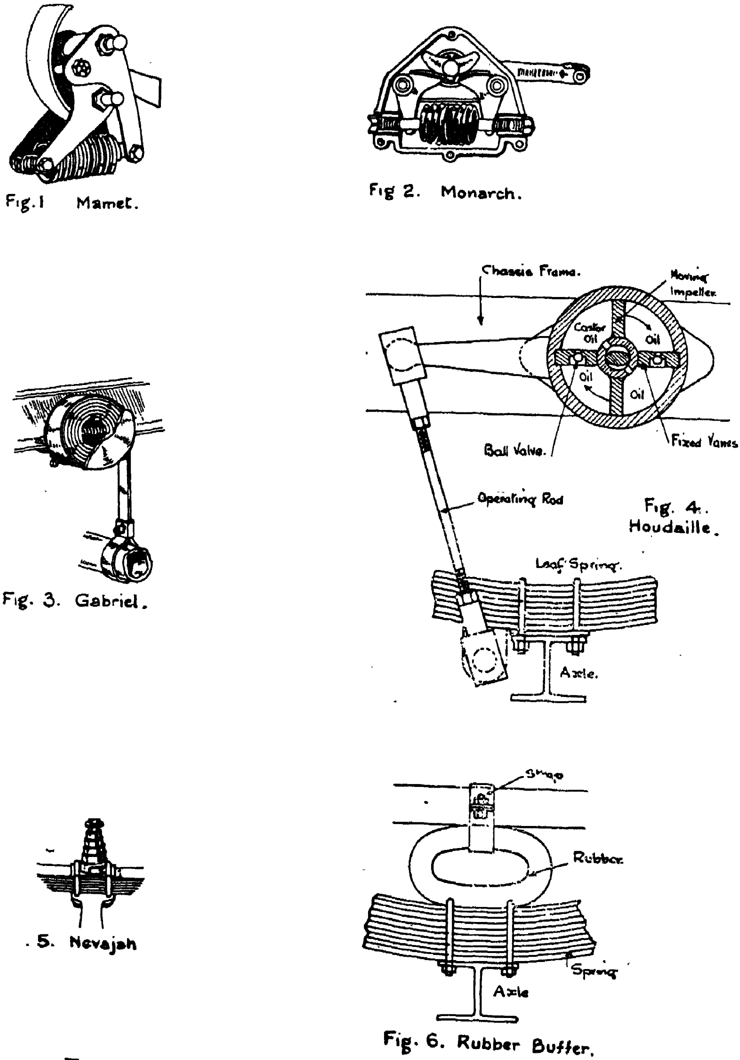


FIG. 56.—Examples of Earlier Type Shock Absorbers.

of the main springs. The hydraulic shock absorber usually consisted of a cylindrical portion attached to the frame, and

a series of vanes which could rotate, or rock in the cylinder, which was filled with oil; a lever attached to the axle of the vanes, and connected by a universally jointed rod to the axle or spring. It was so arranged that when the axle moved upwards, valves opened and allowed the oil to flow freely from one side of the moving vanes to the other so as not to introduce any resistance. When the axle fell the valves closed and the oil trapped could only escape slowly through the small holes provided for the purpose, so that a high resistance was offered; the degree of damping could readily be varied by altering the leakage area. Sometimes rubber balls, or thick annular rings, or even volute springs (as in the Nevajah type) are inserted between the axle and the chassis frame, to act as shock absorbers as shown in Fig. 56 (5) and (6).

Friction Type Shock Absorbers.—The method of damping-out the small movements of the springs by introducing solid frictional devices between the springs and the chassis frame is a very convenient one to employ, since it necessitates simple types of friction dampers. The usual method is to use a pair of flat levers, with a large central friction joint, having Ferodo, or a similar friction material, between steel plates. The outer ends of these levers are pivotally attached to the axle and frame, respectively.

FIG. 57.—André Telecontrol Adjustable Shock Absorber.

It will be seen that any relative movement between the axle and frame is resisted by the friction at the common joint. The amount of this friction is made adjustable—usually by means of a nut-and-screw device.

In some cases a Bowden control is fitted to this friction adjustment, so that the degree of damping can be regulated

from the driver's seat to suit the nature of the load and the road-surface.

The Hartford friction type shock-absorber has two steel arms pivoted together so as to form a cylindrical box, within which is a series of friction discs of specially treated material, held against each other by means of a spring, the pressure of which is adjustable by means of a nut; an indicator is provided to show the relative pressure. The other ends of these arms are pivotally connected to the frame and axle, respectively.

Fig. 57 illustrates an adjustable type of friction damper which is controlled from the driver's seat by means of a Bowden cable; it is known as the André Telecontrol.

In another adjustable model André shock absorber the pressure between the friction surfaces can be controlled hydraulically by means of a dashboard hand-wheel operating a piston for increasing or reducing the oil pressure; in this case the friction surfaces are in the form of shoes which work against the inside of a brake drum. There is an oil gauge on the dashboard to serve as an indicator of the oil pressure in the system.

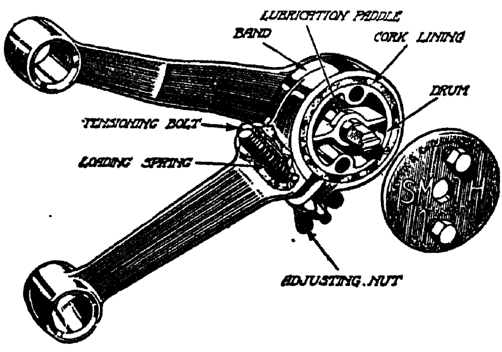


FIG. 58.—Smith Friction Type Shock Absorber.

Another popular friction type of shock-absorber was the Smith one, which has been fitted to several makes of small and medium size cars.

It consists of two arms pivoted together, the extremities being connected to the chassis and axle. The arm connected to the chassis carries a brake-drum, whilst the other arm connected to the axle carries an external brake-band

which can be adjusted to any desired extent by means of a spring-loaded tensioning bolt and wing nut. When the axle rises, the two arms move towards one another; this movement acts on the tension spring thus allowing the band to slide freely round the drum. On the rebound, however, the arms separate and the brake-band coils round the drum, thus absorbing the excess energy stored up in the car springs during the compression movement. In order to allow small spring movements to be uncontrolled by the shock-absorber, a small lever with a screw adjustment is used. This release lever is operated by a pin fixed eccentrically with the drum in such a way as to positively cut out all resistance at low speeds.

It will be seen from the illustration (Fig. 58) that the lever has a recessed bore to house the tensioning spring. The top of the housing is bevelled and the head of tensioning bolt has four indicating grooves so as to enable a uniform adjustment to be made to each absorber.

Luvax Hydraulic Shock Absorber.—The Luvax shock absorber, made by Messrs. J. Lucas Ltd., Birmingham,

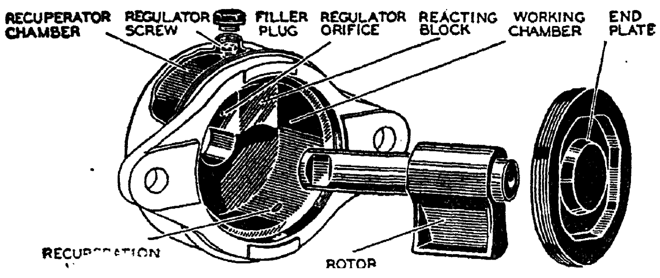


FIG. 59.—Luvax Hydraulic Shock Absorber.

and used on Morris and many other cars, are of the hydraulic type designed to control the main springs both on the bound and recoil. They incorporate pre-set constant pressure valves which prevent the resistance of the shock absorbers rising too far; these valves are accommodated in the reacting block (Fig. 59).

Movement of the shock-absorber spindle which is connected by a lever arm and connecting-rod to the spring, causes the rotor to rock one way or the other in the working chamber which is filled with oil. The rotor experiences a

definite resistance to its motion owing to the pressure of the oil during the latter's displacement in the chamber through the regulator orifice.

To ensure correct operation the working chamber is kept full of oil, automatically. The whole mechanism is arranged for this purpose within another cylindrical chamber, or reservoir, kept nearly full of oil by replenishment every 10,000 miles or so, through a plug provided. Any shortage of oil in the working chamber is made good from this reservoir through valves in the lower wall of the cylinder. Similarly, any air which may be present is expelled through a special air duct in the top of the reaction block; the oil valves seat themselves when the shock absorber is not operating.

In connection with the rotor shaft lever arm and the connecting links between the lever arm and the spring the bearings consist of round blocks of rubber which obviate the use of ball joints and dispense with the necessity of lubrication.

It should be mentioned that Messrs. Lucas Ltd. also make a double-piston opposed hydraulic type of shock absorber somewhat similar in general layout to that shown in Fig. 60; this can be supplied with a variable dash control device to alter the shock absorbing effect at will.

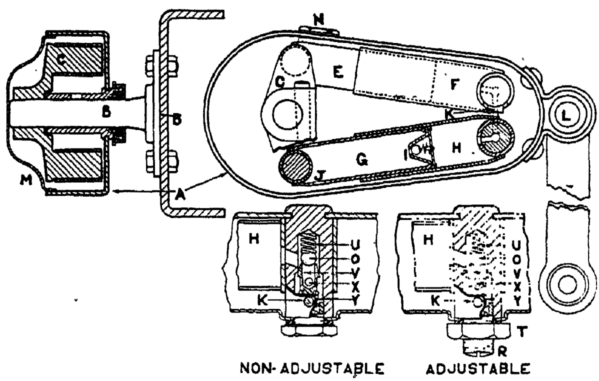


FIG. 60.—Armstrong Shock Absorber.

The Armstrong Shock Absorber.—This compact design of hydraulic shock absorber (Fig. 60) is of the double-acting pattern. It controls both the deflection and rebound

movements of the main springing system and is adjustable to suit the type of car and road conditions. Referring to Fig. 60, there is a double crank *C* which is fixed to the flanged spindle *B*; the latter is mounted on the chassis frame. Two plungers *E* and *G* working in the oil cylinders *F* and *H* are pin-jointed to the ends of the crank *C*. The cylinders, plungers and cranks are all enclosed in a casing *A* which is filled with oil; the latter is introduced through the plug orifice *N*. The casing *A* is connected by means of an arm *L* to the axle of the car by means of a suitable link. Vertical movements of the axle causes oil to be pumped from one cylinder to the other through a ball valve *O* placed in the passage *K*. This valve is controlled by springs *U* one of which is adjustable for tension by means of the screw with lock nut *T* which bears against the spring *Y* in contact with the perforated member *Y* having holes at *X*; in this way the shock absorbing action of the device can be regulated by increasing or reducing the quantity of oil that passes through the valve *O*. Any oil leakage past the plungers is replaced through a hole in the plunger *G* and into *H* through the valve *I*.

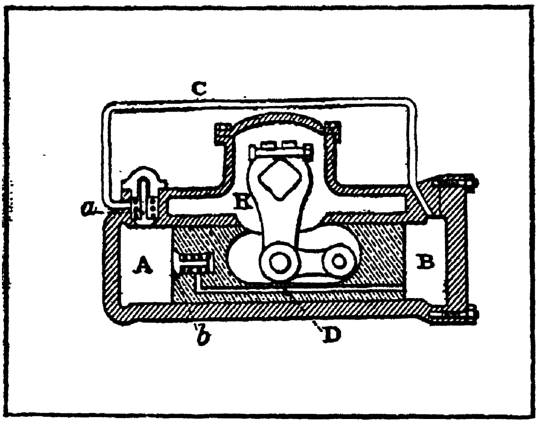


FIG. 61.—Rolls Royce Shock Absorber.

The Rolls-Royce Shock Absorber.—The principle of the shock-absorber that has been used on Rolls Royce cars is illustrated in Fig. 61. The square ended operating arm attached by a vertical connecting-rod having ball-and-socket joints, is connected by means of the simple

linkage to the piston *D*, that works in a cylinder filled with oil at its two ends *A* and *B*. When the operating arm moves as a result of a road shock, oil is displaced from one end of the cylinder to the other past the spring-loaded valves shown at *a* and *b*. In the former case the path is along a passage *C*. The loading of the valves is such that greater resistance is offered to the movement of the piston corresponding to the downward movement of the axle relative to the frame than to the upward movement.

The Delco Shock Absorber.—This double-acting piston type shock absorber is used on several of the popular American cars, including those made by Messrs. General Motors Co. (Fig. 62.)

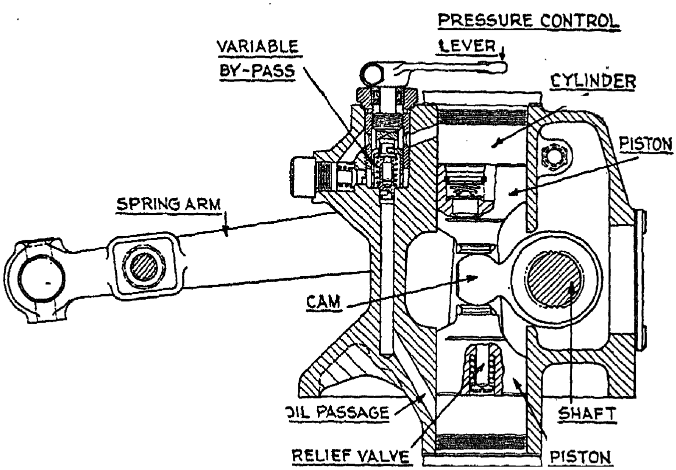


FIG. 62.—Delco Double Piston Shock Absorber Showing Oil Passages and Valves.

In arrangement it is similar to the model previously described in having opposed pistons and cylinders. The unit includes one rebound and one compression cylinder, the pistons being connected together and each piston provided with a spring-loaded inlet valve and a relief valve. The pistons are operated by a cam connected to a shaft outside. The arm on the latter is, in turn, connected to the main spring by means of a connecting-rod having a rubber bushed joint at the top and double rubber block at the bottom or spring end.

Each relief valve has a bleeder hole in the valve stem and is spring-loaded. Under normal car action, when the fluid pressure is applied by the piston, transfer of oil takes place through the bleeder hole; for violent road shocks, however, the relief valve itself opens, thus allowing

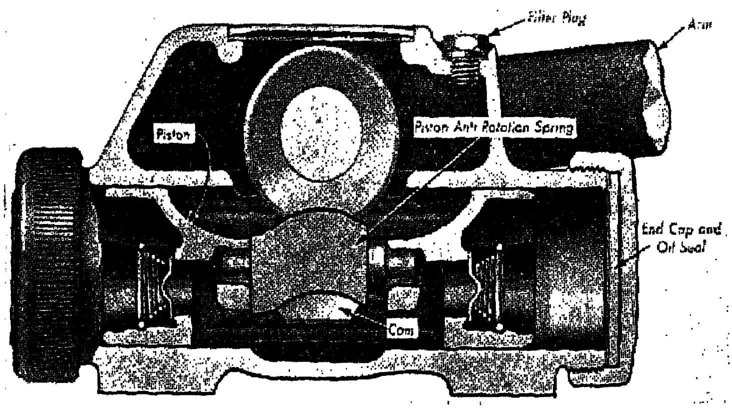


FIG. 63.—Another arrangement of the Delco Shock Absorber.

a more rapid transfer of the oil past the valve seat as well as the bleed hole.

As the shock absorber arm moves downward due to car spring rebound movement, pressure set up in the rebound cylinder forces oil through the bleeder valve, or opens the relief valve by compressing its spring; at the same time the compression inlet valve opens owing to the partial vacuum existing, so that the compression cylinder fills with oil. The action of the oil is the same in both compression and rebound cylinders, but the rate of movement of the shock absorber arm on the compression or upward spring movement is slower on account of its having a smaller size bleeder hole and stronger relief valve spring than in the case of the rebound cylinder.

III

THE FRONT AXLE AND STEERING

The front wheels in the majority of cars run freely, that is to say, they are not positively driven by the engine; except in front drive or 'all-wheel' drive vehicles. The usual arrangement of the latter system is to transmit the engine drive to a special central gearbox, from which four 'propeller' shafts drive the four wheels; each wheel is sprung separately in this case.

The common arrangement adopted is to provide a lateral axle, to carry the front wheels and their steering pivots. This front axle is usually a drop-forging, i.e., is forged in one piece from a good carbon or low nickel steel; a typical example is shown in Fig. 64 (A). In certain small cars a tubular axle was hitherto employed, the *Swivel Pin Head* and *Spring Pads* being separate forgings pinned, or riveted and brazed to the tubular portion as shown in Fig 64 (B). Either of the I-beam or tubular types of section gives a very strong and light axle.

The front axle is provided with special enlarged flanges, one near either end to carry the spring attachments; these flanges are termed the *Spring Pads*.

There is another type of front axle, often used on light cars, and made with a solid, instead of a forked end as shown in Fig. 64 (C). This is a cheaper axle to manufacture, owing to its simpler and straighter design. Its use necessitates a forked *Stub Axle*, and it is not possible to bring the centre of the front wheel as close to the steering pivot centre as in Fig. 64 (A). As we shall see, later, it is necessary to bring the wheel as much as possible over the steering pivot. There are one or two designs which enable this end to be attained. One of these employs a steel disc wheel, domed (or dished) outwards as shown in Fig. 67, and arranges the stub axle member on the top of the front axle steering pivot head; this gives true centre

pivot steering. Modern wire-spoked wheels are now made offset in order to give centre-point steering. Another arrangement that has been used on light cars is shown in Fig. 64 (D).

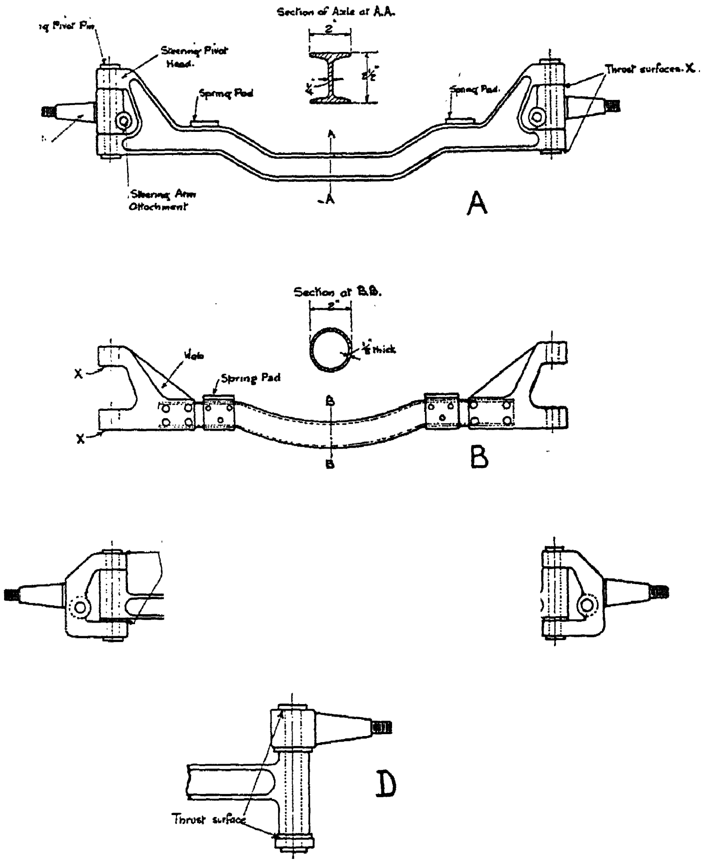


FIG. 64.—Front Axle Arrangements.

Considering the front axle and stub axle, in general, it will be seen that the weight of the front portion of the car is transmitted through the front springs to the front axle, and thence to the stub axles, and finally through the bearings of the latter in the wheels. Since the wheels have to pivot, or turn about the steering pivot pins, and

the weight of the front portion of the car acts between the front and stub axles, it is evident that there will be an endwise thrust on the turning surfaces; very careful design, in the matter of suitable thrust washers, or better still thrust ball races, and in the lubrication arrangements are necessary. In Fig. 64 (A) and (B) the thrust comes on the surfaces marked X; in (C) and (D) the thrust comes on the surfaces shown.

Stub Axle Bearings.—In the case of rigid front axles, the usual arrangement is to employ stub axles having forked ends which are bushed in order to provide bearings for the king pins. The latter are made drive fits in their holes in the axle ends and are secured in position by means

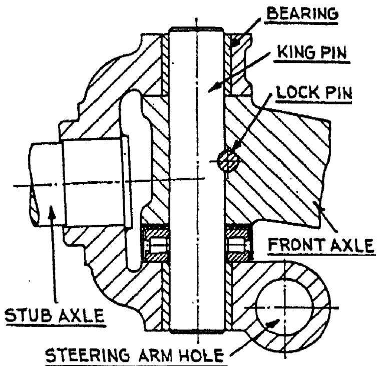


FIG. 65.—Stub Axle Bearings.

of locking pins driven through holes in the sides of the axle holes; these pins engage with corresponding grooves in the central parts of the king pins. Fig 65 shows the components of the stub axle bearings.

The King Pin Bearings.—The thrust on the king-pin bearing may be taken on a plain, ball or roller bearing, the two latter methods being preferable from the view point of greatly reduced friction. Fig. 66 shows one method of taking the car weight thrust by means of a special thrust roller bearing which gives line contact instead of the usual point contact of the ball-bearing. The upper and lower king pin bearings are phosphor bronze bushes, lubricated by means of grease forced under pressure

through the grease-gun nipple on top and down [illegible] ough the hole drilled in the pin.

The wheel shown in Fig. 66 is of the detachable [illegible] wire-spoked type, the hub being a push-fit on to the axle [illegible] boss; the wheel is attached to the latter by four studs and [illegible] nuts.

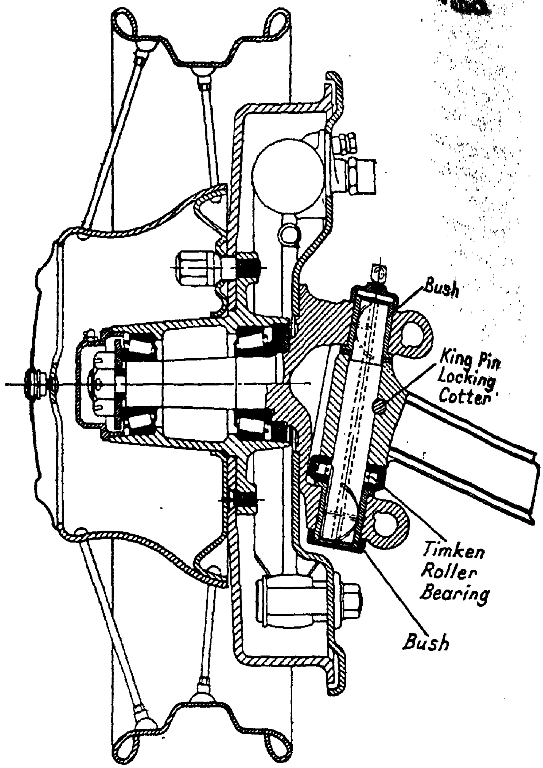


FIG. 66.—Front Wheel and King Pin Bearings.

Timken taper roller bearings are fitted to the stub-axle, the arrangement shown taking both the direct loads and end thrusts; provision is also made for taking up wear in the bearings by moving the conical housings in an endwise direction.

The steering swivel-pin is inclined in the axle end, so as to bring the continuation of its axis to the point of contact of the centre of the tyre with the ground; this arrangement ensures easier steering.

Front Brake Axle Design.—With the general adoption of front-wheel brakes, the additional braking stresses incurred by the front axle have necessitated a general stiffening up of this member. The portion between the spring pads and the stub-axle is now made of circular, or I-beam section, in order to take the torsional stresses due to the brakes. Fig. 68 shows a good design of front axle for front-wheel brakes. It will be seen that the central portion between the spring pads is of I-beam section, whilst the outer overhung portions are of circular section, tapering slightly towards the swivel-pin

Another front axle arrangement is shown in Fig. 69. In this case the axle is of I-beam form throughout and the overhung portions are upswept towards the swivel-pin ends. This illustration shows also the hydraulic shock-absorbers for the springing system and the steering connections.

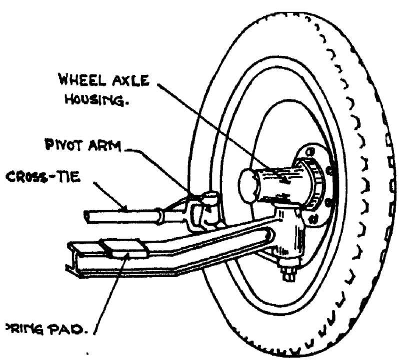


FIG 67.—True Centre Point Steering

The Steering Arm.—Before leaving this part of the subject, attention should be drawn, in Fig. 64 (A), to the steering arm attachment on the stub axle.

This steering arm is only fitted to one of the two stub axles, namely to the left, in English and the right, in American and Continental practice, as viewed from the front of the car.

This steering arm and its hardened steel ball end can be moved in a fore and aft direction by the steering mechanism of the car, so that the stub axle can be rotated

about the steering pivot pin; if the ball end is moved towards the front of the car (or the reader viewing Fig. 64 A), the stub axle will move towards the rear, and since both front wheels are connected so as to move in the same

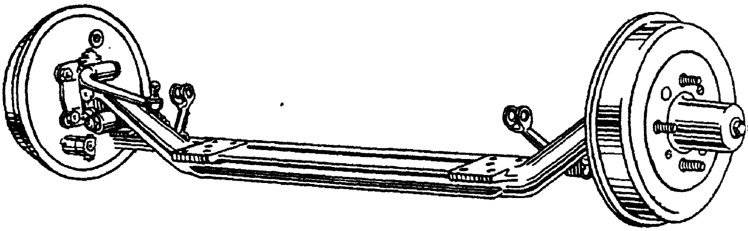


FIG. 68.—Front Axle Designed for Taking Braking Stresses.

direction, the car will steer to the driver's right; similarly a movement of the steering arm backwards will result in a left hand turn, as viewed from the driver's seat.

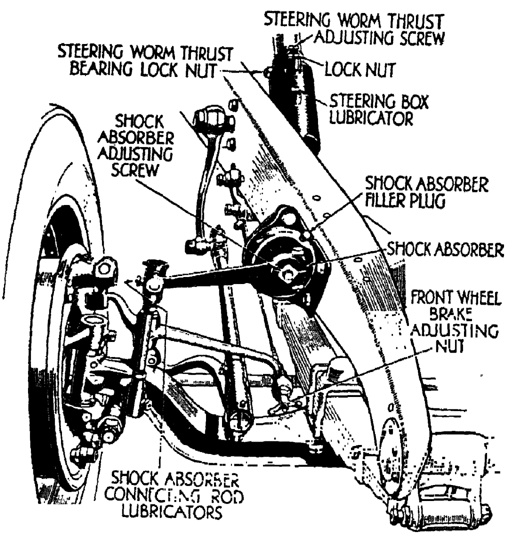


FIG. 69.—Typical Front Axle, showing Steering Mechanism and Hydraulic Shock Absorber.

The Drag Link.—This connecting-rod between the drop-arm of the steering gear box and the steering arm on the stub axle is provided with ball-and-socket joints

at each end. Usually, the drag link is of tubular construction and carries a pair of divided ball sockets at each end. One half socket is fixed and the other is held against

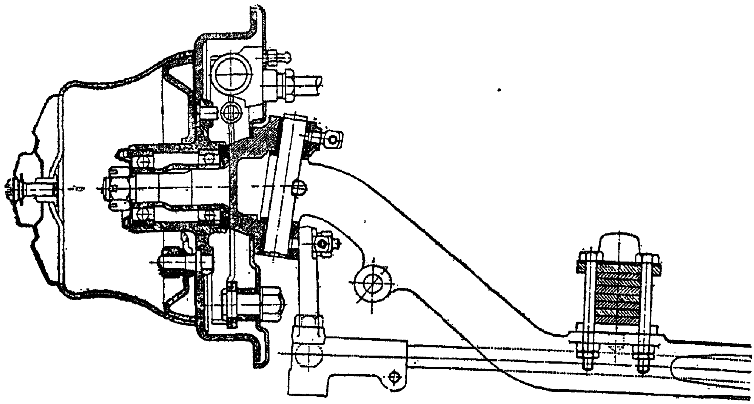


FIG. 70.—Front Axle with Ball Bearings for the Wheel and Hub.

the ball end, of the drop or steering arm, by means of a strong compression spring, thus giving an arrangement for taking up wear and also providing a kind of shock absorber. Usually a screwed end plug is fitted in order to vary the compression of the spring and to take up wear effects; this plug is made with castellated end for locking with a split pin.

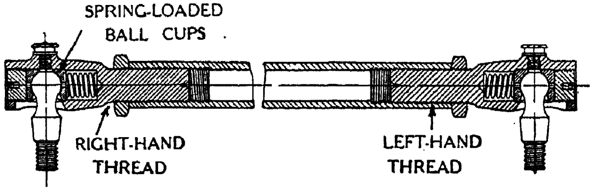


FIG. 71.—The Drag Link or Link Rod.

Alternative Drag Link End Bearings. More recently alternative designs of drag link bearings to the simple ball-and-socket joints have been introduced. In the case of the Ross design the drag link end casing is filled with $\frac{1}{8}$ in. and $\frac{3}{32}$ in. steel balls. In order to provide for the initial adjustment a spherical-ended set screw is used to project through the casing. This type of low friction joint has been shown,

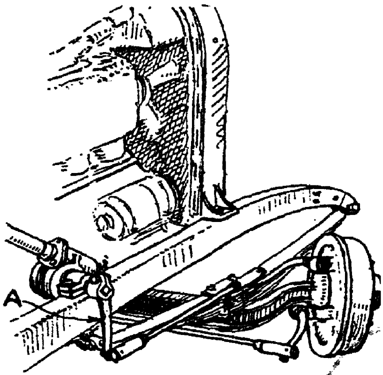


FIG. 72.

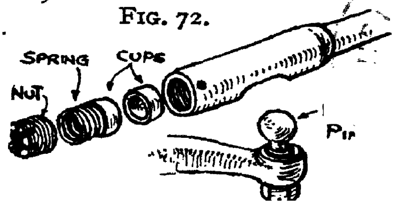


FIG. 72.—(Above) Showing Steering
Link in Position.
FIG. 73.—(Below) Steering Drag Link
ponents.

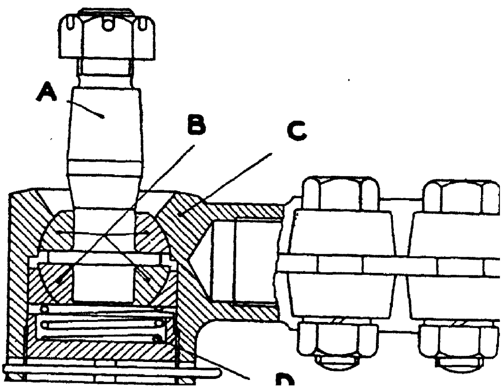


FIG. 74.—Section Through Ball Joint, showing how the Fou
Phenolic Moulded Segments are applied and preloade
(*A*) Steel Pin ; (*B*) Four Bakelite Segments Formin
Ball ; (*C*) Steel Housing with Chrome-Plated Interior
(*D*) Cover Plate

by endurance tests, to give no measurable wear or backlash after very long periods of service.

The use of synthetic resin materials, such as Bakelite, for the steering joints is another recent innovation (Fig. 74). When first tried, the spherical recess surface rusted and caused the plastic surface to wear rapidly. This difficulty was overcome by chrome plating the surfaces of the recess, but this resulted in squeaking or 'groaning' when the joint was moved. To avoid this trouble, the segments were moulded from a shock-resisting type of material using a fabric filler (presumably finely shredded) and loaded with colloidal graphite, which provides sufficient lubrication. It was found necessary, however, to load the bearing surfaces to about 100-lb. per sq. in. to secure prolonged service without deterioration. It is stated that the moulded parts require no machining before assembly.

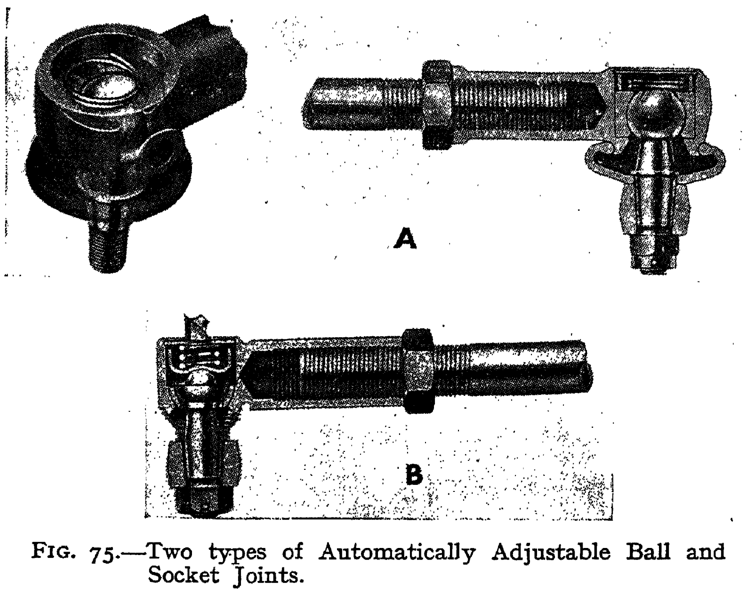


FIG. 75.—Two types of Automatically Adjustable Ball and Socket Joints.

An improved type of ball-and-socket bearing for drag links and also tie rods, known as the Thompson Eccentric Joint, illustrated in Figs. 75 (A) and 75 (B) possesses certain advantages over the orthodox pattern. Its chief merit is the automatic taking up of wear, whilst maintaining the

steering free from rattle and road shock
joint is provided with a spring device whi
contact between the ball and its divided e
members so as to take up wear effects. No provision is therefore provided for adjustment and the ball and socket unit is permanently sealed by means of a swaged in disc (above). Grease gun lubrication is arranged for, a nipple being provided for this purpose. A permanent "rubber-boot" enclosure on the other side prevents the entry of water, mud or dirt.

Fig. 75 (A) shows the normal pattern drag-link joint which allows for angular movements up to 40°.

The type shown in Fig. 75 (B) is designed for cars having independent springing with relatively short coupling rods which necessitate angular movement between the ball pin and rod which may be as much as five times that required in the normal leaf-springing system. The same principle of taking up wear effects, automatically, is employed and the ball joint has a floating bush between the ball pin and the socket to provide for the increased movement. A different spring controlled laminated conical sealing device is used to exclude water, etc.

These drag-link units are becoming widely used and are a product of the Lockheed Company.

The Steering Mechanism.—We have seen that the steering of the car is accomplished by moving the steering arm ball backwards and forwards in a fore and aft direction, and that the other wheel is connected to the operated one so as to move with it. Now the steering gear mechanism, which is usually housed in a casing attached to the chassis frame is simply a means for moving this steel ball end as required. Each stub axle has a lever, or arm attached to it, and the two arms are connected by a rod running across, and under the frame, pin bearings being provided at either end; this connecting-rod is termed the *Cross Tie Rod* or *Track Rod;* it is rigid in construction, and is provided with a threaded portion and nuts, for altering its length; this is necessary for adjusting the parallelism, or alignment of the front wheels.

Fig. 76 illustrates the principle employed in modern steering arrangements. Referring to Fig. 76 (B), the *Steering Wheel* rotates, at the other end of the *Steering*

column, a *Worm* which engages with a *Worm Wheel Sector.* On the same shaft is attached a lever known as the *Drop Arm ;* this is rocked to and fro by the worm wheel, and in so moving causes the ball end and the steering arm Fig. 76 (A) to move in a fore-and-aft direction. Referring to Fig. (C), it will be seen that as the left hand stub axle is

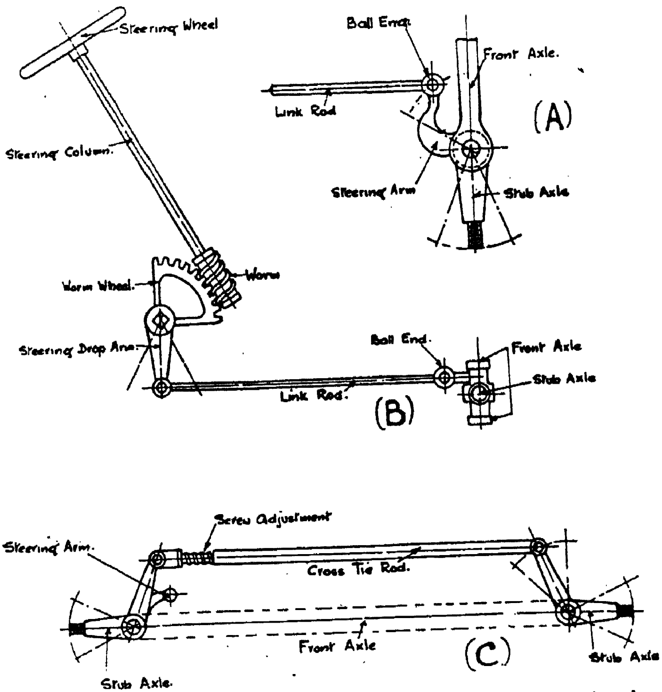


FIG. 76.—Principle of the Ackermann Steering Mechanism.

rotated, it communicates its motion through the *Cross-tie* or *Track Rod* to the right hand stub axle, so that both wheels move together.

It should be noted, that as the ends of the drop lever and the steering arm are ball-jointed, this allows the front wheels and axle to move up and down vertically under the action of the springing, without placing a drag on the steering.

As regards the direction of steering, if the steering wheel is turned in a clockwise manner as viewed from the driver's

seat the car steers to the right; if anti-clockwise, to the left.

A complete steering gear layout is given in Fig. 77 for the Morris cars. The various parts of the system are clearly defined and alternative names for the 'link-rod' and 'cross-tie' rod are given in the illustration.

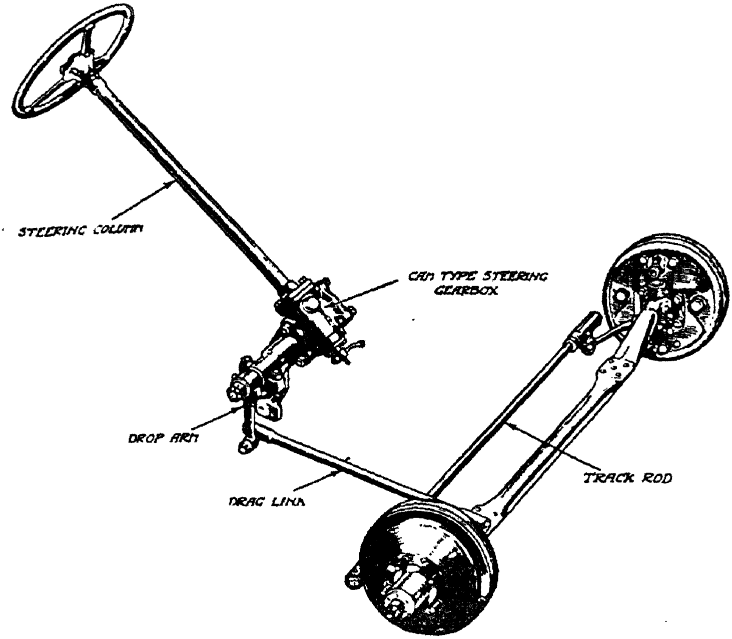


FIG. 77.—The Morris Steering Mechanism.

Steering Gears.—There are several methods of operating the steering drop arm, by the movements of the steering wheel, of which the most common systems are (1) The Worm and Nut; (2) Worm and Worm Wheel; (3) The Cam, and (4) Epicyclic or Direct Gears. In the case of a few of the earlier cars, until recently on the roads, a direct method was used, whereby there was an inclined steering column having a lever arm at its lower end, which was connected directly, by means of a ball-jointed connecting link rod to the steering arm. The disadvantage of *direct steering* lies in the fact that all road shocks on the wheels are transmitted directly to the steering wheel, and that the front wheels are constantly tending to turn the steering

this type of steering is also known as *Reversible*
An *Irreversible Steering* system is such that no
the road wheels either by hand, or due to lateral
will tend to move the steering wheel. Such a
however, results in a loss of steering 'feel,' on the part of the driver, which experience has shown to be undesirable; moreover the rigidity of such a system results in increased wear in the drop and steering arm joints. The present tendency is to compromise matters by using *Semi-Reversible* steering; in this case there is a slight tendency for the road wheels, if turned, to move the steering wheel, but owing to the spring joints in the steering links there is sufficient elasticity to prevent road shock effects from being felt; there is also just sufficient force required at the wheel, to give the desirable steering 'feel.'

Another important item is the *Steering Ratio*, that is, the relation between the movement of the steering wheel and that of the front wheels. In slow speed and commercial vehicles the steering wheel has a relatively large movement for a small one of the front wheel pivots; this enables the driver to exert a larger turning effort on the wheels, but the steering movement at the wheels is slower. For fast cars it is desirable to have a more delicate steering control, for a small movement of the steering arms at high speeds means quite a large lateral swerve. It is, therefore,
arrange for a larger movement of the steering
for a given steering wheel movement in such cases. The total amount of steering arm angular movement determines the greatest amount of front wheel turning which can be obtained. When a steering gear, or mechanism is so designed that the front wheels have the maximum sidewise inclination or turn, when the steering wheel is turned as far as it will go in one direction, it is said to have a good '*Steering Lock.*' The greater the amount of this lock, for a given wheel-base, the smaller will be the *Turning Circle* of the car. In the case of small cars the diameter of the turning circle usually lies between 25 and 35 feet. Taxicabs are given good steering lock, in order that they may turn in the smallest circles; with their relatively short wheel bases they can easily turn in an ordinary 25 ft. wide road. The turning circles of large touring cars range from 35 to 45 ft. diameter.

Steering Ratios.—In recent years, owing to the necessity of accurate control of the car at higher road speeds there has been a tendency for the steering ratio to increase; the most favoured value is between 3 and 4 complete turns of the wheel from lock to lock; the greater ratio has also been necessitated by considerations of large section low pressure tyres. A disadvantage of a high steering ratio is the longer time required to turn the car at low speeds and in reverse.

Worm and Nut Steering Gear.—In this method, which is commonly used on both small and large cars, the steering shaft ends in a square-cut screw thread, which in the simplest

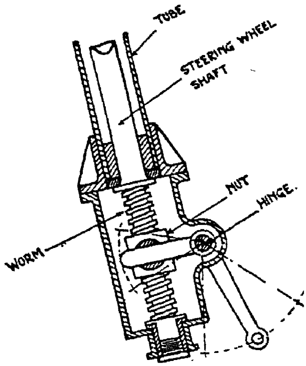


FIG. 78.—Worm and Nut Type of Steering Gear.

forms works in a nut; the operation of turning the steering wheel moves this nut along the steering rod screw thread, and the nut actuates a bell-crank lever, pivoted as shown in Fig. 78 in the steering box casing. The steering-gear drop arm forms the other arm of this bell-crank.

The Armstrong Siddeley steering gear belongs to the worm and nut class, in which the nut has a trunnion pin bearing on either side in the rocking arm member. Referring to Fig. 79 the worm is shown at *A*; the trunnion oil nipple at *B*; steering box casting at *C*; steering lock adjusting screws at *D*; the nut engaging with the worm at *E* and the trunnions at *F*. It should be mentioned that the shaft near *F* is the one to which the drop arm is rigidly secured. The amount of movement of the rocker shaft *EF* is limited by the set screws shown

the amount of wheel lock can thereby be

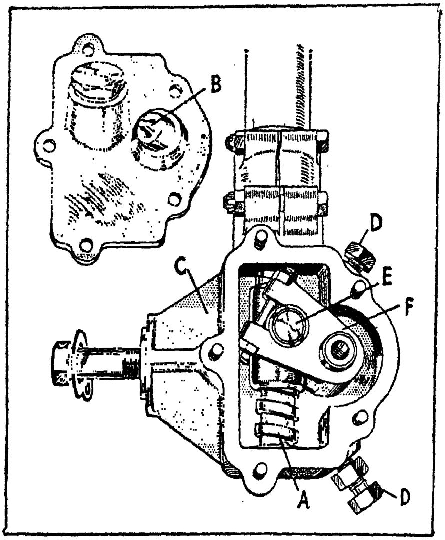


FIG. 79.—Armstrong-Siddeley Worm and Nut Steering Gear Box.

An elaboration of the worm and nut principle is shown in ig. 80; this is employed on the more expensive cars. In

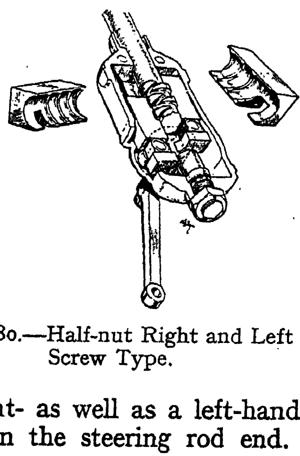


FIG. 80.—Half-nut Right and Left Hand Screw Type.

this case, a right- as well as a left-handed square screw thread is cut on the steering rod end. The respective

threads engage each with a half-nut, so that when the shaft is rotated one half moves up, say, and the other down. Suitably cut slots in the half-nuts engage with square members free to rotate upon pins located at the two ends of a lever pivoted about its centre. An extension on this lever forms the drop-arm member. It will be seen that the operation of turning the hand-wheel moves one square member up, and the other down, thus causing the Tee-shaped drop-arm lever to rotate.

In some cases the two square threads are formed on different parts of the steering rod end, the half-nuts being then displaced, axially. It is necessary to provide a ball-thrust with gears of this type, to take the end thrust of the worm.

The Worm and Worm Wheel.—Here, there is a square thread, or worm, on the steering rod end, as before, but instead of working in a nut, or half-nuts, it engages with a worm-wheel as shown in Fig. 81. The drop-arm is

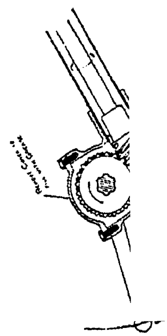


FIG. 81.—Worm and Complete Worm Wheel Steering Gear

keyed to the same shaft as the worm wheel, and works rigidly with it. Usually a squared or castellated shaft is used for the worm wheel, so that as wear of the worm sector, in action, occurs, the worm wheel can be turned round to a new position, and the drop-arm taken off, and replaced in its correct working position. The arc of movement of the drop arm is usually from 60° to 90°. Many makers provide only a sector of a worm-wheel for this reason.

this gives a smaller and lighter mechanism, but has nc
rovision for wear (Fig. 83).

The principal components of a typical worm and wheel

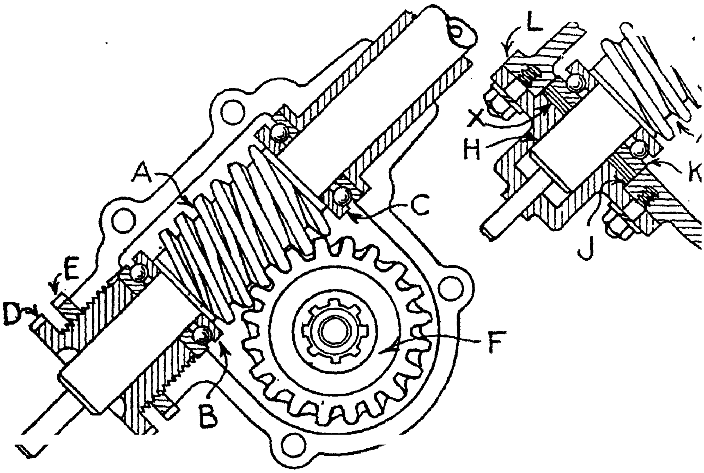


FIG. 82.—Worm and Wheel Steering Gear.

steering gear unit are shown in Fig. 82. Referring to t
left hand illustration the worm A is secured to the steeri

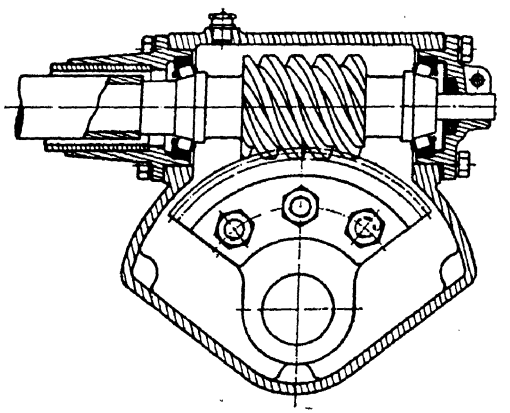


FIG. 83.—Worm and Sector Steering Gear.

olumn shaft and has thrust bearings at B and C
spectively. It meshes with the worm wheel F

Provision for adjusting or taking up the wear effects in the thrust bearings is made by the adjusting screw member D, which has a lock nut E.

An alternative method used for taking up end play is shown in the right hand illustration. This consists of an end cap plate member which forms a spigot bearing for the steering column shaft H ; it is bolted to the casing of the steering gear box L and has a spigot J. Between the face X of the cap member and the outer race of the thrust bearing K a number of thin liners, or shims, of varying thicknesses are arranged. When it is necessary to take up end play of the thrust washer one or more shims of the required thickness (to equal the amount of the end play) are removed and the end cap again bolted securely in place.

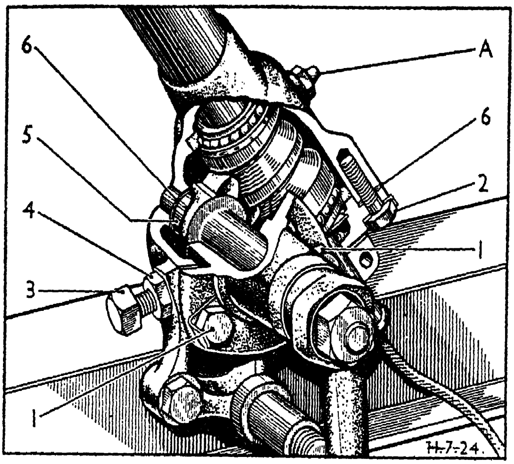


FIG. 84.—The Austin Steering Box.
1, 3 and 4, Mesh Adjustments. 2, End Cover Nuts
Thrust Button. 6, Shims. ...e.

Hour-Glass Steering Gear.— ... gear used
e Austin small cars is of th ...
large-toothed
ing gear box

End play adjustment of the steering cross shaft consists in introducing one or more shims 6, behind the thrust button 5.

Adjustment of the mesh of the worm and sector is

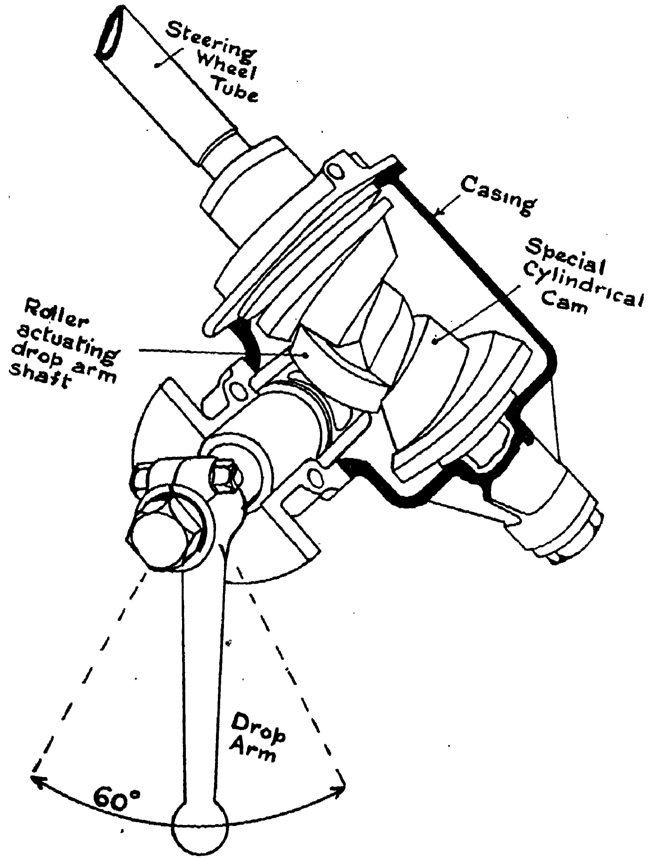


FIG. 85.—Marles Type of Cam and Roller Steering Gear.

obtained by first loosening the three nuts 1 and locknut 4, afterwards turning the screw 3 in a clockwise direction to take up the slack.

The worm is provided with ball journal and thrust bearings (Fig. 84) and is arranged so that there is minimum

backlash in the normal, or straight-ahead position, the backlash increasing towards the full lock.

Cam Type of Steering.—In this case the drop arm forms one arm of a bell-crank lever, the other end carrying a ball-bearing roller which is moved up and down the steering rod, by means of a special type of cam surface (Fig. 85), somewhat like a hollow distorted thread of coarse pitch. This cam-screw always touches the roller at two diametrically opposite places, so that there is no play. The long cylindrical portion in Fig. 85 represents the bearing about which the bell-crank rotates; the pin carrying the roller is shown. The Marles steering gear belongs to this type; in one model the steering wheel moves through $1\frac{3}{4}$ revolutions for the complete 60° movement of the drop-arm; the pitch of the cam is kept constant. This gives a very free steering, without backlash or shock-transmission.

The Marles double-roller pattern steering gear, shown in Figs. 86, 87 and 88, possesses certain advantages over the

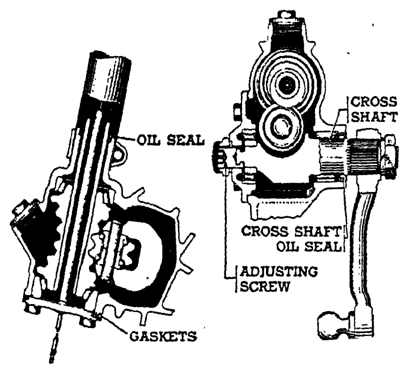


FIG. 86.—Double Roller Cam Steering Gear.

original design illustrated in Fig. 85. The worm or cam is of hardened alloy steel and engages with a twin-section roller, which is also of hardened steel provided with needle bearings to rotate upon a fixed pin carried in one end of the rocker shaft; the latter is mounted on needle bearings in the steering gear box casing (Fig. 88). It has also a supporting bearing on the left and a screw adjustment is provided for

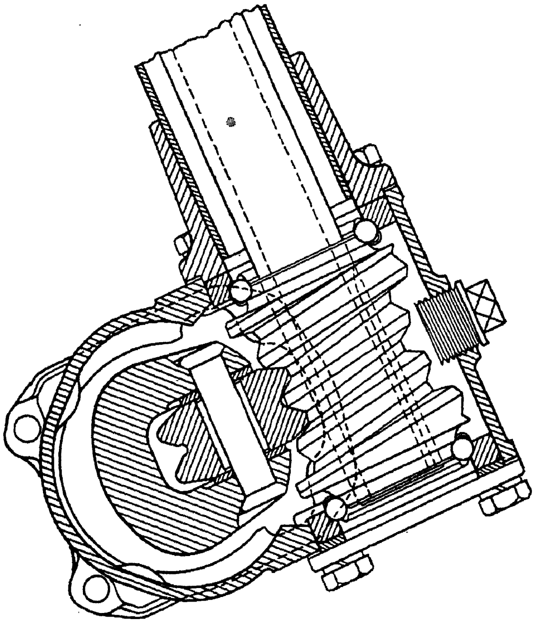


FIG. 87.—Double Roller Type Marles Steering Gear, showing Worm or Cam.

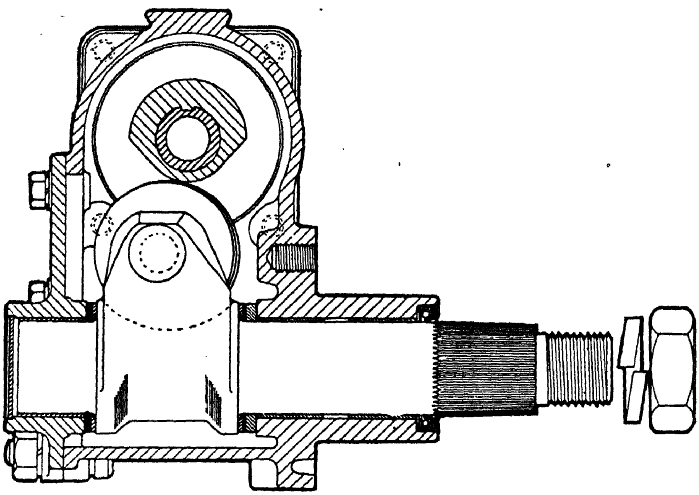


FIG. 88.—Rocker Shaft of Double Roller Type Marles Steering Gear

end location. The cam member is mounted either on adjustable tapered roller bearings, as shown in Fig. 86, or upon heavy pattern ball bearings designed to take both thrust and journal loads (Fig. 87). The use of needle bearings to the roller and rocker shaft enables a very compact design of gear box to be used. An alternative type of steering gear box is provided with trunnion bearings on each side of the roller support portion; this design is suitable for heavy vehicles.

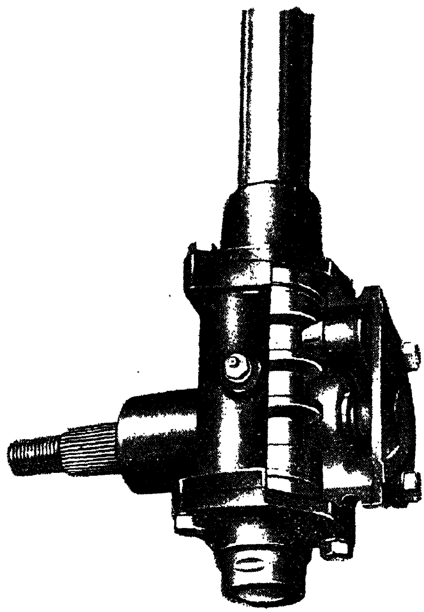


FIG. 89.—The Bishop Steering Gear.

The Bishop Steering Gear.—Another modern type of steering gear used on British cars is that known as the Bishop type (Fig. 89). In this case there is a worm connected to the steering wheel tube, so as to turn with the steering wheel. A conical roller of the same section as the hollow between the teeth of the worm is moved in a direction up and down the axis of the worm when the latter is rotated. The roller in question is mounted at the end of

an arm attached to the drop-lever shaft; the threaded and splined end of this shaft is shown on the left, in Fig. 89.

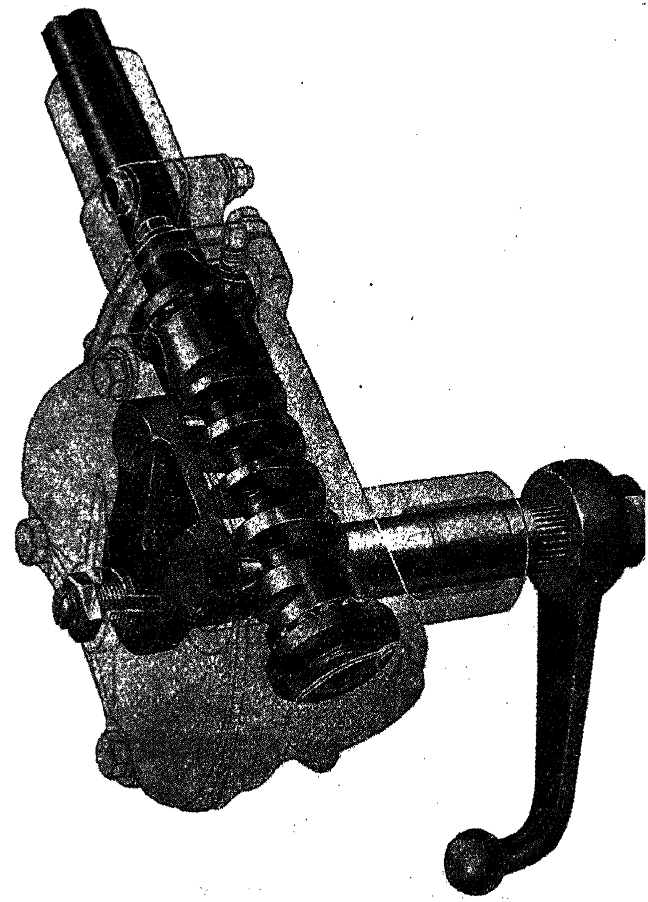


FIG. 90.—The Ross Twin Lever Steering Gea

The Ross Twin Lever Steering Gear. This resemble in principle the Bishop gear but employs twin conica rollers on the rocking member. This design enables greater mechanical advantage or steering gear ratio to b obtained over the whole range of steering angle, the rati actually increasing at the greater angles, as shown i

Fig. 91. The dotted line shows the gear ratios for the single lever Ross steering gear; it will be observed that within the 15° (or parking range) the mechanical advantage is almost doubled so that only about one-half the physical effort is needed for the twin lever type gear.

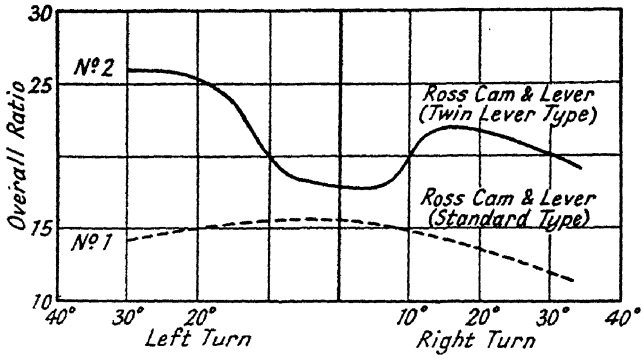


FIG. 91.—Performance Curves for Single and Twin Lever Ross Steering Gears.

of angular motion is 100° as compared with the normal 80° for the single lever pattern gear. The increased contact area due to the use of two rollers reduces the frequency of adjustment for wear effects by about 33 per cent.

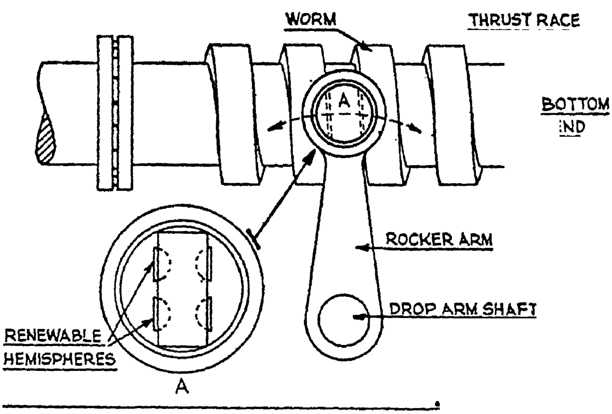


FIG. 92.—The Marles-Weller Steering Gear.

The Marles-Weller Steering Gear.—In this design a special worm or cam-like spiral groove transmits the neces-

sary rocking movement to the rocker arm by means of a parallel flat-sided member *A* (Fig. 92) which engages between adjacent faces of the helical cam ; the short shaft carrying this member can rock in its hole in the end of the rocker arm. In order to assist alignment of the faces of *A* on the helical cam faces special hemispherical members are fitted, as shown in the lower left hand sketch ; an additional advantage is that should wear occur, new hemispheres can be fitted.

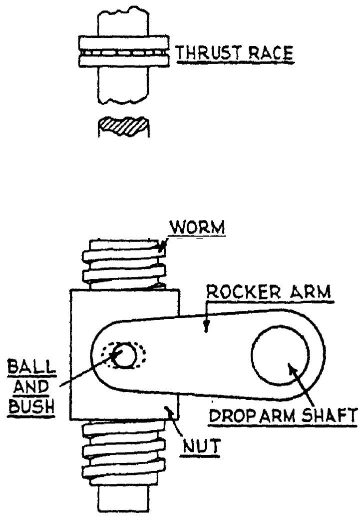


FIG. 93.—The Burman-Douglas Steering Gear.

The Burman-Douglas Steering Gear.—The principle of this gear is shown in Fig. 93. In this type a square threaded worm at the lower end of the steering column engages with a nut having on one side a ball and bush device for transmitting the movement of the nut to the rocker arm.

Continuous Ball Type Steering Gear. A method of reducing the friction between the nut member and the worm in the case of the worm and sector pattern steering gear used on Cadillac cars is illustrated in Fig. 94. In this design an endless 'chain' of balls is employed between the

worm and nut members, the nut forming a kind of rack ving axial movement so as to cause the sector

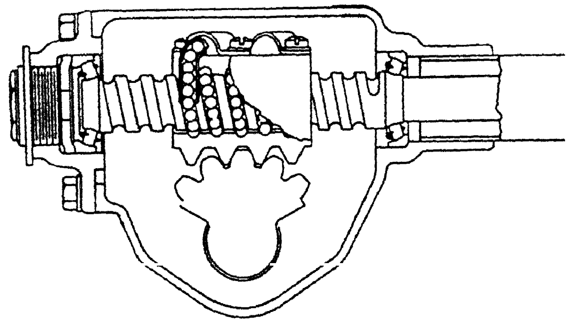


FIG. 94.—Continuous Ball Bearing Type of Steering Gear.

rocker arm to rotate or rock. The balls roll continuousl between the worm and nut, return chambers being provide between the ends and centre of the worm.

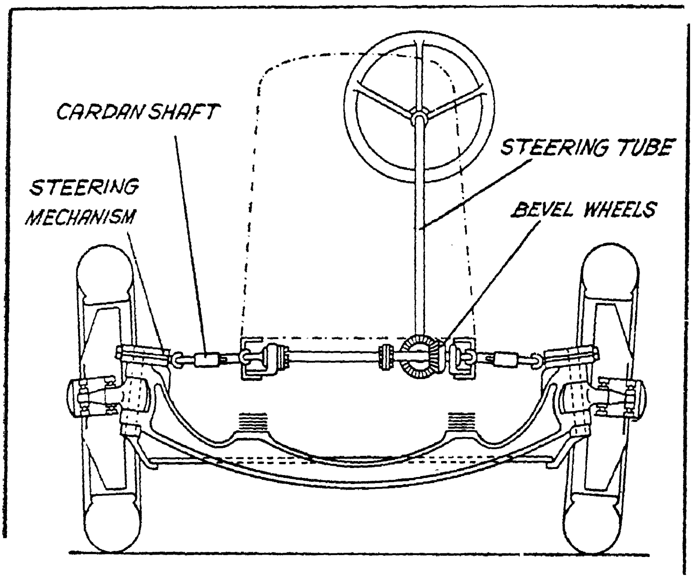


FIG. 95.—Positive Steering to Both Wheels.

Positive Steering to Both Wheels.—In view of fact that there have been several accidents in the past

to failures of the steering system, certain firms now fit positive steering to both front wheels.

The ordinary steering system applies the steering effort from the steering wheel to one of the front wheels only, so that if anything happens to the drop-arm or drag-link, or their connections, the car may get out of control.

It is not difficult, however, to apply a positive drive to each of the front wheels, although the application of such a drive introduces a little more complication and cost to the car. Incidentally, most independently sprung front drive cars have positive steering systems.

Fig. 95 shows one method, adopted by a French car, in which a universally jointed cross-shaft is rotated by a pair of bevel wheels, one on the steering column and the other

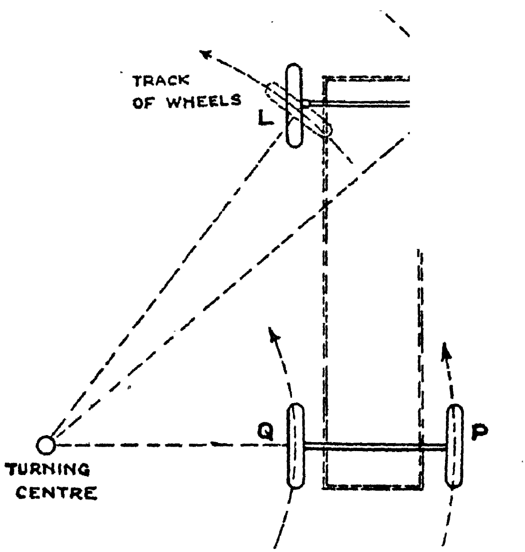


FIG. 96.—Instantaneous Centre of Turning.

Some Theoretical Considerations on Steering.—The
s of a car must be turned in a definite manner
relation to each o , ar s of the
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eferring t

mechanical law that for pure turning of the four wheels, they must always rotate about the one centre; the latter need not be fixed, however, but may change as the front wheels turn. This centre is termed the *Instantaneous Centre.* Evidently, since the rear axis is fixed it must be somewhere on the extension of the axis *PQ.* If now the dotted lines *OL* and *OR* be drawn at right angles to the dotted positions *L* and *R* of the front wheels, these lines, for true steering must meet the axis *PQ* produced in *O*; the resultant path of the wheels is indicated by the arrows.

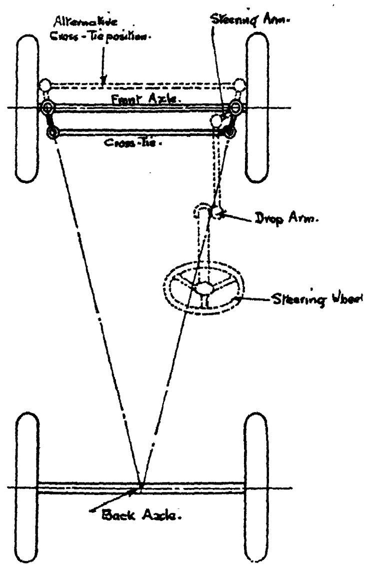


FIG. 97.—Ackermann Principle of Steering.

The *Ackermann System* of steering, shown in Fig. 97, gives, very nearly, this method of front wheel steering. The cross-tie may either be in front of, or behind the front axle, provided the steering arms are disposed as shown, i.e., splayed outwards when in front, or inwards when behind; usually the steering arms when produced meet on the back axle. It will be evident, upon consideration that the instantaneous centre *O* is at an infinite distance when the front wheels are straight, but as they are turned progressively it approaches the car. The cross-tie is in

compression, when behind, and in tension when in front of the rear axle. An important point to remember in regard to the Ackermann method of steering is that when the front wheels are steered round a corner *the rear wheels do not follow the same paths* but describe a path of flatter curvature, thus ' cutting across ' the corner, as it were.

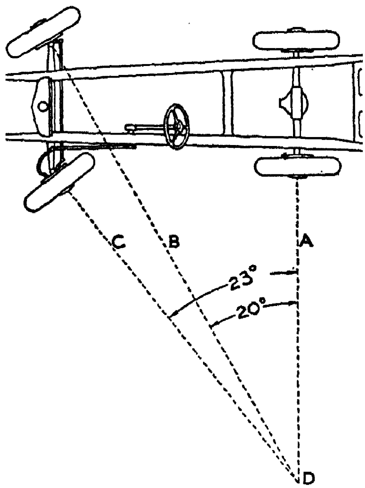


FIG. 98.—Showing the ' Full Lock ' Angles of a Typical Car.

For this reason it is necessary to take a somewhat wider sweep with the front wheels when turning corners; similarly, when reversing a car the rear wheels tend to turn outwards more than the front ones.

Castor Steering is the name given to the steering effect obtained when the steering pivot or ' king-pin ' axis is inclined so that when produced, it meets the ground ahead of the point of contact as shown in Fig. 99 (A); the distance *AB* is known as the *trail;* the angle *ACB* is usually from 3° to 5°. The object of castor steering is to give the front wheels a trailing or castoring effect so that they tend to return to the straight after a turn. With castor steering, it is inadvisable to reverse the car

at more than a slow speed, or the steering will tend to 'take charge'. The castor angle of the system is usually altered

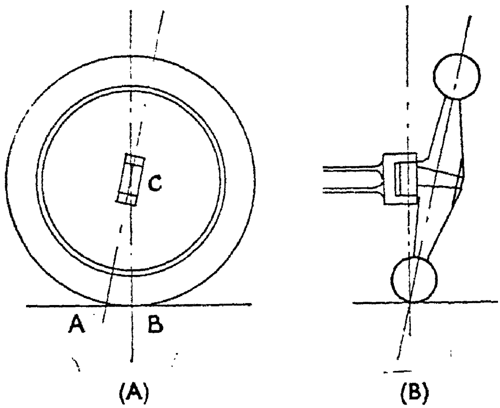


FIG. 99.—The Castor Type of Steering (A), and the method of inclining the stub axles to reduce steering drag (B).

by means of a thin steel wedge between the axle spring pad and the lower spring blade.

Camber Angle. In addition to providing for the castoring action the front wheels are usually inclined

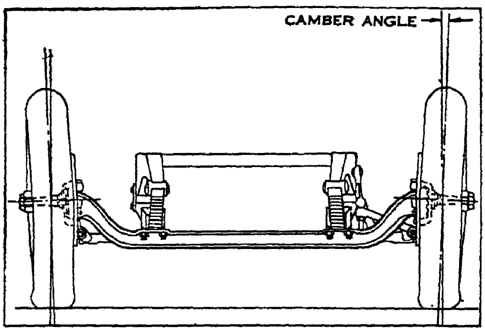


FIG. 100.—Illustrating Wheel Camber.

inwards at their lower ends as viewed from the front of the car (Fig. 100). The object of this inclination is to bring the centre of the tyre—where it contacts with the ground—approximately under the axis of the steering

pin in order to reduce the steering drag which otherwise would be experienced. Thus, if the central plane of the tyre were set outwards from the steering pin axis there would be an appreciable torque on the stub axle tending to make each wheel turn outwards at the front end.

The ideal method of steering is that known as 'Center Point,' in which the king-pin axis produced meets the centre point of contact of the tyre with the ground, as indicated in the arrangement shown in Fig. 99 (B). It is possible with either the pressed steel or wire-spoked type of wheel to arrange for the axle end and king-pin to lie within the outer plane of the wheel in order to give centre-point steering, without the aids of camber or king-pin angle.

The camber angle should be such that the contact point of the tyre centre is directly at the intersection of the king-pin axis and the horizontal ground. The angle in question is usually from $\frac{1}{2}°$ to $1\frac{1}{2}°$. If the wheel has too much camber not only will there be tyre drag but also excessive tyre wear on the outer edges of the tread, since the tyre will be forced into a conical shape on its lower side. Insufficient camber will cause steering drag and a tendency for the tyres to wear more on the inner edges of their treads.

King-Pin Angle. In the case of the smaller diameters of wheels the camber angle or set of the stub axle axis

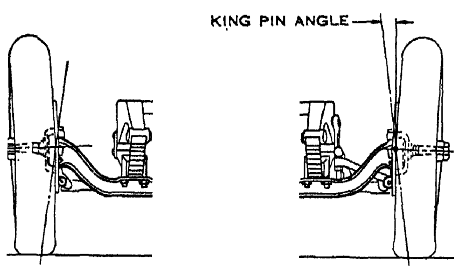


Fig. 101.—Inclination of King Pin.

may become excessive, so that a compromise is usually effected by inclining the king-pin axis (Fig. 101) so as to reduce the camber angle; if there is no camber angle the

king-pin angle must be increased to bring the pin axis to the central contact of the tyre. The usual king-pin angle is from 5° to 8°.

'Toe-in' of The Wheels.—In addition to 'castor' and 'camber' the front wheels of practically all cars are inclined slightly towards each other at the front in order to prevent excessive tyre wear and to take up the spring of the steering mechanism.

The amount of 'toe-in' of a pair of wheels is the difference between the two measurements *B* and *A* (Fig. 102).

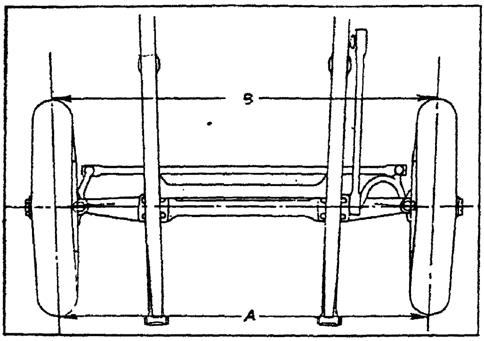


FIG. 102.—Wheel 'Toe-in.'

Just as the object of camber is to give the wheel a position such that it will be almost balanced from the point of view of steering effort, so with 'toe-in' the purpose is to set the wheel in a position to reduce to a minimum the road friction on the tyre. The usual amount of 'toe-in' varies from about $\frac{1}{8}$ in. to $\frac{3}{8}$ in. according to the design of chassis; this adjustment is made by means of the screwed portion of the track rod. For small cars up to about 12 h.p. the usual 'toe-in' is from $\frac{1}{8}$ in. to $\frac{3}{16}$ in.

Independently Sprung Wheel Steering.—The design of the steering mechanism requires careful consideration when the front wheels are sprung independently, for as previously mentioned there may otherwise be an interaction between the springing and steering systems.

The manner in which the front wheels move under the springing action must be taken into consideration for, if the wheel moves forwards or backwards the steering may be affected adversely.

It is now a common practice to operate the steering in the manner shown in Fig. 103 using a divided track rod

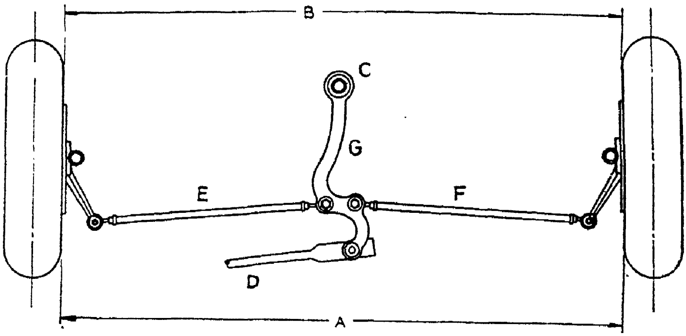


FIG. 103.—Positive Steering as used for Independently Sprung Front Wheels.

or equivalent rods *E* and *F*, operated laterally by means of the lever *D* connected to the steering gear box drop arm. The lever *G* has a fixed bearing at *C* so that it merely

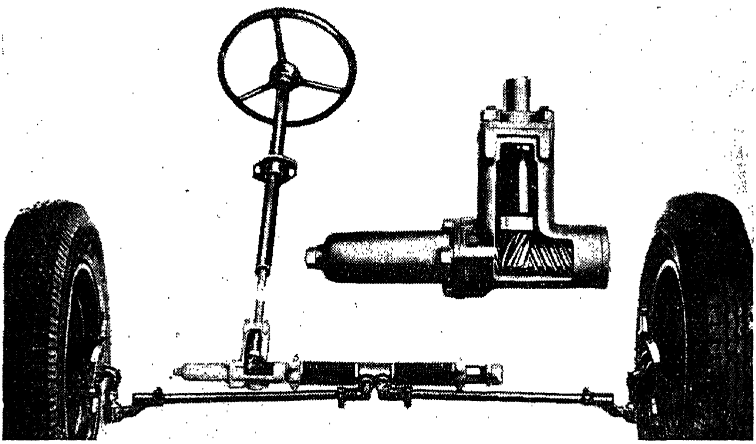


FIG. 104.—Citroen Front Wheel Steering System.

rocks to and fro when the car is steered. A method of operating the divided track-rod used on the Citroen front drive cars (Fig. 104) consists in attaching the inner ends of the two rods to a sliding bar having at one end a

spiral gear engaging with another one attached to the lower end of the steering column; this form of direct steering is both simple and efficient.

Power Steering.—In view of the increased physical effort necessary to steer the larger sizes of motor vehicles when fitted with large section low-to-medium pressure tyres, attention has been given to the possibilities of assisting the driver by utilizing a vacuum or air pressure operated Servo unit. A typical layout for such a system, viz., the Feeny and Johnson one, is shown in Fig. 105.

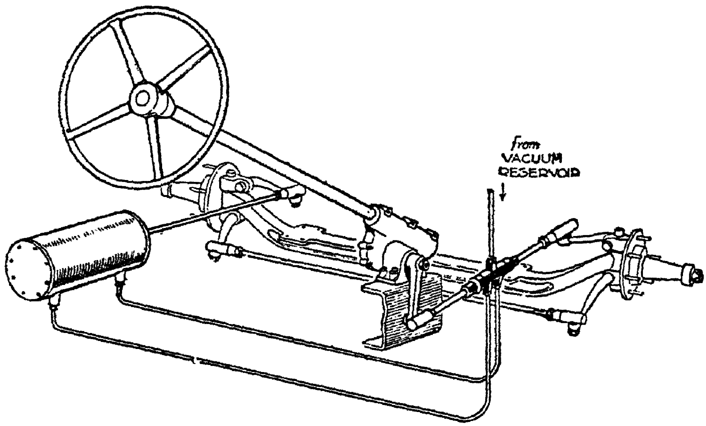


FIG. 105—A Method of Power Steering employed on Large Cars and Commercial Vehicles.

In this example there is a double-acting valve, coupled direct to the steering gear box. The valve unit has a central connection with the vacuum source or reservoir and two outlets connected by pipes to either end of a vacuum cylinder; the latter has a piston, the rod of which is connected to the left-hand steering arm. When the steering wheel is moved clockwise it not only operates the drag link, but also moves the vacuum valve so as to put the rear end of the Servo cylinder into communication with the source of vacuum supply. The Servo piston, under atmospheric influence—for the front of piston is placed into direct communication with the air by the same movement of the vacuum valve—then moves backwards thus pulling the rod and left hand steering arm

backwards. This movement is transmitted by the track rod to the right hand steering arm, so that the Servo unit supplies power control to both wheel steering arms.

Another design of Servo steering mechanism due to the Dewandre firm is shown, schematically, in Fig. 106. It

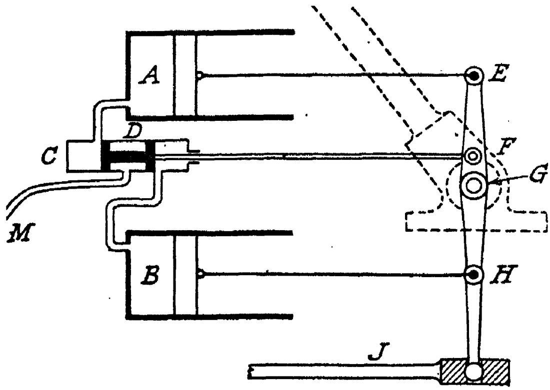


FIG. 106.—Dewandre Power Steering Method.

comprises two single acting vacuum power cylinders A and B connected by tension rods to the ends E and H of a lever keyed to the rocking shaft of the steering gear box G. A vacuum control valve D, in a cylinder C, is connected by a rod to the previously mentioned lever at F. When the driver commences to turn his steering wheel so as to move the rocking shaft lever upper part E to the left hand, the valve D is moved to the left, so that the vacuum in the central space of valve D, which is communicated from the inlet manifold *via* the pipe M, is applied to the cylinder A. Atmospheric pressure on the piston of this cylinder then forces it to the left, thus applying a pull through the tension rod to the end E of the lever in the left hand direction. For right hand movement of E, the lower cylinder B comes into action.

An improvement on this method is to operate the valve D by a short arm on the rocker shaft G, and to mount the lever E H on a bearing over the rocker shaft extension so as to act independently of the steering gear itself. The lower end of the lever EH has a ball end and engages with the steering connecting rod J.

Wheel Wobble.—This effect consists of a fairly rapid sideways oscillation of the front wheels when the steering wheel is held rigidly. It may be caused by slack in the steering mechanism joints, or elasticity in one or more of the steering or spring members. A common cause is the moving backwards of one of the shackled ends of the front springs which causes the front axle to tilt or deflect rearwards on that side. If the steering connecting rod is too short, it will also tend to move in a small circular arc about the

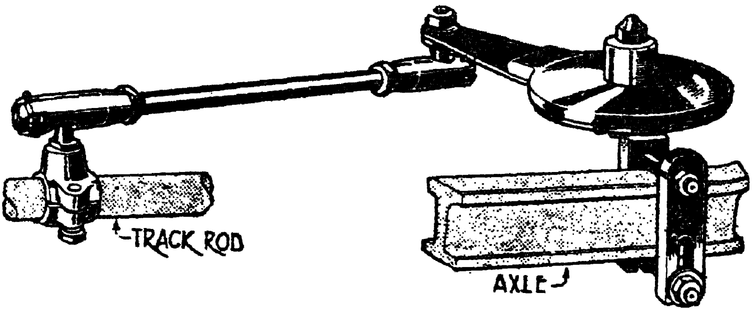


FIG. 107.—The F.E.W. Damping Device for Preventing Wheel Wobble.

drop-arm ball joint, whilst the front axle will move more or less vertically; the net result will be that when one wheel strikes a bump the front wheels will turn slightly inwards, and if they drop into a depression, slightly outwards. The effect can be minimised by making the connecting rod as long as possible. Cars fitted with quarter elliptic and also with transverse springing are very prone to show steering wobble, due to incorrect alignment of the springs, or wear in the radius rod joints, in the latter type of springing.

Fig. 107* illustrates one method of curing wheel wobble, by the introduction of a friction damping device between the front axle and the track-rod.

The F.E.W. damper shown consists of a ball member clamped securely to the track-rod, and a connecting-rod having sockets at each end to connect it with the end of a flat metal arm which can rotate about a pin attached to the axle. A circular disc of friction fabric is clamped between the other end of the flat arm and a disc secured to the axle as shown. The movement of the flat arm is always,

* Reproduced by permission of *The Motor*.

therefore, resisted by the friction of the fabric disc; this effectively damps out any lateral vibrations.

Care and Attention of the Steering Gear.—Most steering wheels will show a little backlash, if tested with the car stationary. It is perhaps a good plan to have a small amount of play, as it enables the driver to hold the wheel in such a way that any inadvertent small movement will not cause the car to deviate from its course; it also helps to absorb road shocks. The usual permissible amount of play is about 3° to 5°, or from $\frac{3}{8}$ ins. to $\frac{5}{8}$ ins. on a 14 inch steering wheel. Most steering gear boxes are provided with adjustments in the form of adjustable screws or 'shims' to take up *end play in the steering rod worm* or thread—which is the usual cause of backlash. The ball-joints on the connecting-link, drop-arm and steering arms should be kept well-greased, and enclosed in soft leather covers to keep out grit and wet.

The steering gearbox must be lubricated occasionally with suitable oil or grease, and the lubrication of the top of the column under the steering wheel should not be overlooked. The cross-tie joints must be lubricated at brief intervals; it is better to enclose them in leather covers. Grease lubricators are always fitted to the steering pivot pins and their thrust-washers; in cases of 'heavy' steering, traceable to the pivot pins, jack up both wheels so as to take the weight of the car off the pivots, and screw down the greasers, working the wheels to and fro at the same time.

It is very important to remember that *the front wheels should not be turned by the steering wheel when the car is at rest;* the mechanism may become severely stressed; in one or two cases brought to our notice the cross-tie has actually been bent. If the front wheels are pulled or pushed by someone at the same time as the hand wheel is moved, this risk will be obviated.

The Steering Wheel Controls.—In order to render the control of the car as simple as possible it is the practice to place the engine and electric controls on the steering wheel, so that the driver need only disengage one hand to operate these; some cars, however, still have the headlamp switches on the dashboard.

A common plan is to mount the electric horn button in the centre of the steering wheel, *i.e.*, at the top of the steering column. Mounted concentrically with this is a graduated disc, or cap, with suitable thumb-levers for the

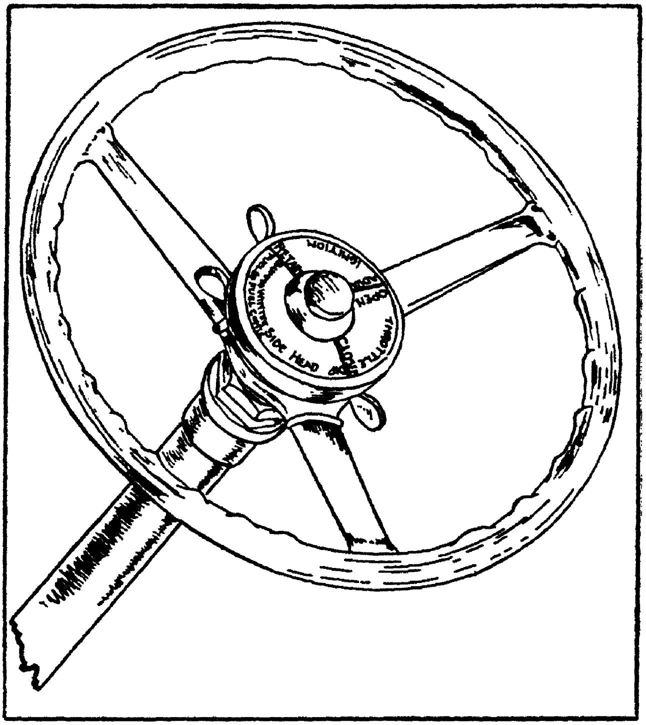


FIG. 108.—Typical Steering Wheel Controls.

throttle hand control, the ignition advance and the electric light switching, respectively.

In some cases there are three concentric caps, the centre one for the horn, the middle one for the hand throttle control and thè outer one for the electric lights. By rotating these caps—which are suitably serrated for the purpose—the throttle and electric controls are operated. In this case there is no ignition control, as an automatic spark advance is fitted in the contact breaker casing. This grouping of the principal controls is known as "*Finger Tip Control.*"

Improved Steering Wheel Designs.—The ordinary three- or four-spoked cellulose covered rigid steering wheel which has been in use for some considerable time has been replaced on certain cars by a wheel having spring-pattern spokes in order to afford a certain degree of flexibility in the direction of the steering column without, however, introducing any freedom of movement in the steering directions.

A more recent development is the Safety Grip Spinner Wheel (Fig. 109) used on certain American cars, which

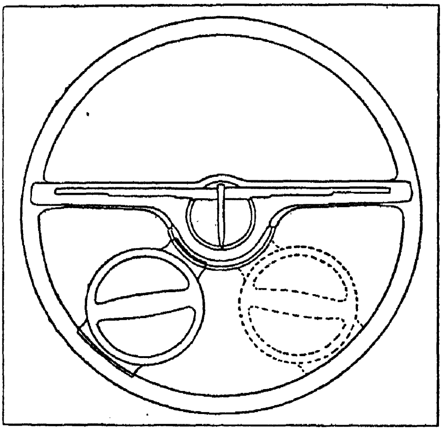


FIG. 109.—Improved Design of Steering Wheel.

enables a firm, constant and safe grip under both straight-ahead and turning conditions, there being no intermediate spokes to obstruct the driver's action. The smaller circle illustrates the spinner grip device which can be adjusted to the most suitable position to suit the individual driver—as indicated by the dotted line.

Instrument Board and Controls.—The usual arrangement on modern cars is to place the instruments on the dashboard, the control pedals on the floor, convenient for the driver, and the rest of the controls on the steering wheel.

Engine Instruments and Controls.—The engine instruments generally provided on most cars include the *Oil Pressure Gauge, Throttle Hand Control, Mixture Control*

or *Air Choke.* In some cases also a hand set ignition advance over-riding control is included for setting the zero of the automatic ignition advance to suit the grade of fuel used, or the state of carbon formation in the cylinder heads; as carbon forms so the ignition must gradually be retarded in order to avoid 'knocking'.

The air choke control, unless automatically-operated by a thermostatic device, is mounted on the dashboard.

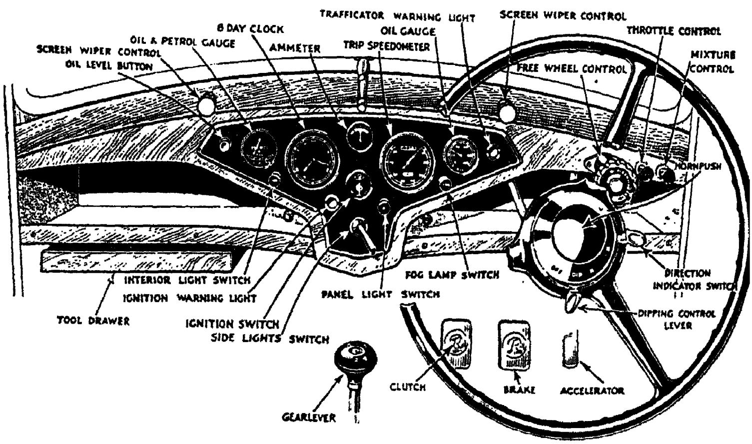


FIG. 110 —Instruments and Controls of the Rover Car.

It should be of the hand held out type such that after the engine has started and is idling properly, it can be released and returned to its full-open position by spring action; this obviates any trouble due to running the engine on the road with the choke closed.

In many cases a cylinder head or radiator thermometer is fitted as an indication of the temperature of the cooling water. With the increasing use of thermostats in the cooling system the custom of fitting these thermometers has diminished; nevertheless, it is always a good feature to have such a thermometer for use as an indication of overheating and for showing when the engine has attained its correct operating temperature.

Petrol and sometimes oil level gauges are also fitted on modern cars. In the case of sporting cars and certain others of the more expensive make, engine rev-counters are included in the dashboard instruments since the

readings of such speedometers afford useful information for gear changing speeds, performance, etc.

The usual arrangement adopted for the foot controls of British cars is to have the *Clutch Pedal* on the left, the *Foot Brake Pedal* in the centre, and the *Accelerator Pedal* on the right.

Electrical Instruments and Controls, etc.—The electrical instruments and fitments include the *Ammeter, Ignition Switch, Head, Side and Tail Lamp Switches, Panel Illumination Switch, Windscreen Wiper (Electrical) Switch, Ignition Warning Red Light, Trafficator Switch* (and sometimes warning lamp), *Horn Button* and *Engine Starting Motor Switch.* The latter may be on the dash-board or, more conveniently on the floorboard operated by a pedal. In cases where dipping headlights are employed it is usual to operate the dipping or ' dip and switch ' control by means of a pedal-type switch on the floor board.

Other electrical devices occasionally fitted include battery level indicators, battery capacity gauges ; windscreen electrically heated de-frosters ; small electric motor driven fan heaters or ventilators for the interiors of closed type cars ; electric clocks ; car radio sets ; engine oil heaters ; cigarette lighters, etc.

Miscellaneous Devices.—Whilst it is desirable to reduce the number of instruments and controls to a minimum, in order to simplify the driving of cars, some of the more expensive cars have special fittings or refinements. These include controls for adjusting the shock-absorbers from the driver's seat, hand adjusters for the brakes, seat position and steering wheel position and inclination adjusters.

When the car has a free-wheel device it is invariably provided with a locking control operated by the driver to put the free-wheel in or out of action.

Cars fitted with pre-selective and other remote control gear boxes have small gear levers and sectors mounted on the steering column, under the steering wheel, for preselecting the gear to be changed into ; a special foot change pedal is also provided for pre-selective gear boxes.

CHAPTER

THE CLUTCH

Its Purpose.—It is necessary when driving a car to be able to connect or disconnect the engine crankshaft and the gearbox driving shaft. In the first place it is impossible to change the meshing of any of the gears in the gearbox, whilst any power is being transmitted through them; as we shall learn in the next chapter, the gearbox contains several trains of wheels which are often engaged by means of sliding dog-clutches. Secondly, it is often necessary to suddenly disconnect the engine drive from the transmission, in emergency. Finally, the different gears in the gearbox can only be meshed by disconnecting the engine drive from the gear shafts and speeding up one or other.

The mechanical device which is fitted between the engine and the gearbox for these purposes is termed the *Clutch.* It consists, broadly speaking, of two members, one fixed securely to the flywheel or crankshaft of the engine, so as to rotate with them, and the other mounted on a keyed shaft connected with the gearbox, in such a way that whilst always driving the main gear-shaft, it can slide along it and frictionally engage with the engine driven member. Fig. 111 illustrates the simplest type of clutch, known as the cone clutch, a brief account of which will assist in explaining the function of all clutches. The cone-shaped metal member *A* is usually the engine flywheel also. The light metal cone *B* has an outer conical-shaped covering of a suitable friction material —usually leather or fabric. It is normally pressed tightly into engagement with *A* by means of a central compression spring *S*, or by a series of small springs around its outer surface. The engine therefore drives the shaft *C*, purely by the frictional grip between the conical surfaces. Now the cone *B*, and its central boss can slide along the shaft

C, but is forced to rotate with it, owing to the key on *C*, and keyway *F* in the boss of *B*. In order to disconnect *B* from *A*, a pedal (known as the *Clutch-Pedal*) *P* is provided. This pedal is pivoted at *D*, and its extension engages with the collar *E* on the clutch boss. By pressing the clutch pedal in the direction shown by the arrow, *B* is disconnected from *A*. Although we have described a very simple type, it should be stated that all types of clutches work on exactly the same principle, namely of frictional engagement between two surfaces, or sets of surfaces, one set on the engine crankshaft extension, and the other on the gearbox shaft and capable of sliding on the latter for disengagement purposes.

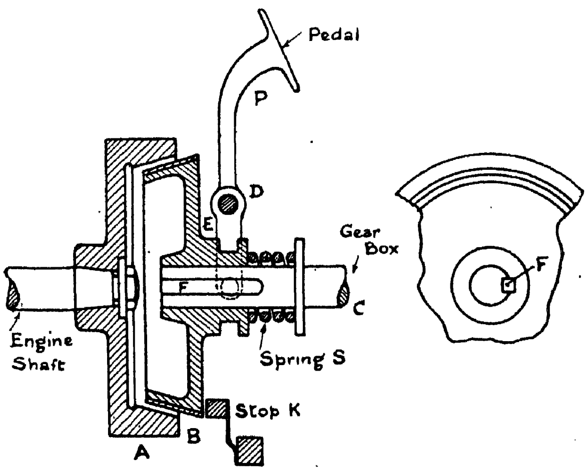


FIG. 111.—The Simple Cone Clutch.

Types of Clutches.—There are four chief types of automobile clutches that are or have been employed for cars as follows: (1) *The Direct Cone*; (2) *The Inverted Cone**; (3) *The Single Plate*, and (4) *The Multiple Plate*. Hitherto the first two types practically monopolised motor cars. More recently the single and multiple fabric and plate types have come to the fore and entirely ousted the former types.

It is desirable that any type of automobile clutch shall possess the following qualities: (1) Ease of operation, that

* Described in 3rd Edition of this Volume.

is to say, it should require a minimum of physical effort to disengage; (2) Freedom from slipping when engaged—early leather cone clutches were bad offenders in this respect; (3) Long life of the wearing surfaces; (4) Absence of noise or backlash in operation; (5) Simple adjustment means; (6) Easy accessibility; (7) Proper lubrication means for the metal working parts (preferably automatic)

The single plate clutch has, to a considerable extent, replaced the cone clutch in most countries, for it does not suffer from the drawbacks of the latter, is entirely enclosed as a rule, is foolproof, and has a very long useful life. Moreover, as the spinning member can be made very light, gear-changing is simplified.

Clutch and Brake Lining Materials.—The friction materials used for clutches include leather, cork, cotton fabric and asbestos-base materials. The two latter materials suitably treated are also used for motor car brake linings. Leather has a (dry) coefficient of friction of 0.27 on iron; the effect of oil is to reduce its value to about one-half. Cork, on iron or steel has a (dry) coefficient of friction of 0.32, this value falling to about 0.18 when lubricated. Fabric friction materials suitably woven, impregnated with certain liquids and compressed, hydraulically, have frictional coefficients of 0.4 to 0.5 (dry). They are not used, however, in cases where much heat is liable to be generated.

Asbestos in the white natural state has a frictional coefficient, on steel, of 0.6 to 0.75, but the effect of binders is to reduce this value by about one-third. It is usual to employ brass wire-woven, impregnated asbestos for braking materials, a frictional coefficient of 0.3 being obtained.

Ferodo and Raybestos are examples of excellent asbestos base friction materials used for clutches and brakes. Raybestos has a coefficient of friction of 0.35 to 0.40 for temperatures up to about 130° C. An energy absorption of 24,000 to 30,000 ft. lbs. per sq. in. of lining is obtainable, with long life, provided the temperature does not exceed 300° C.

The usual clutch pressures for dry plate clutches are from 20 to 30 lbs. per sq. in., but pressures up to 45 lbs. per sq. in. can be used where good cooling of the plates is

ensured. For leather clutches the maximum allowable pressure is 7 lbs. per sq. in., and the maximum speed 3,500 to 4,000 ft. per min.

Clutch Material Temperature and Wear. In a well-designed clutch the dry plate pattern the faces of the driving and driven members should come together gradually with little pressure at first; the rubbing velocity is initially a maximum. As the pressure between the plates increases the rubbing velocity is reduced progressively. The amount of rubbing or slipping should be a minimum under the higher plate pressures since otherwise the surfaces will wear. If the plates slip under load conditions they heat up quickly and since frictional materials are affected by heat, the rate of wear increases rapidly with temperature rise. In the design of plate clutches some consideration must therefore be given to ventilation or cooling of the plates; the use of perforated or flexible plates facilitates clutch cooling, by allowing the air to circulate over the surfaces when the driven plate rotates.

Plate Clutches.—In these a number of parallel discs of metal and friction material are arranged to transmit the drive. These clutches are exceedingly efficient, silent and long wearing, requiring little attention when in use. The simplest form is the *Single Plate* clutch, in which a fabric or metal disc takes the place of the inner member of the cone-clutch; it can rotate with and slide along the gear-shaft member. When the clutch is engaged this disc is gripped between two flat metal surfaces forming part of the flywheel member, and the drive is thus transmitted to the gearbox shaft. The principal components of a single plate clutch are shown in Fig. 112. The clutch pedal, with hinged foot pad, carries on the other side of its bearing a short lever *R*, such that when the clutch pedal is pushed from right to left the end of *R* moves the thrust collar *Q* to the left. The opposite face of *Q* then engages with the upper end of a lever *M* having a pin-joint at its lower end. There are three levers, spaced, at 120° apart, similar to *M*. A screwed stud *N*, locked by nuts *O* in the lever *M* has a bearing on the end of the stud *P*, secured to the flywheel *C*. Thus when *M* is pushed towards the left *N* pushes on the stud *P* (which may be regarded as a fixed member) so

that the plate *J*—known as the *Pressure Plate*—is forced to the right. Normally the presser plate *J* is forced against the friction plate *F* by means of a series of peripheral springs *K*; usually six equally spaced springs are provided so as to give a uniform pressure. These springs are located by other studs, such as *L*.

When the clutch pedal is depressed it has to overcome

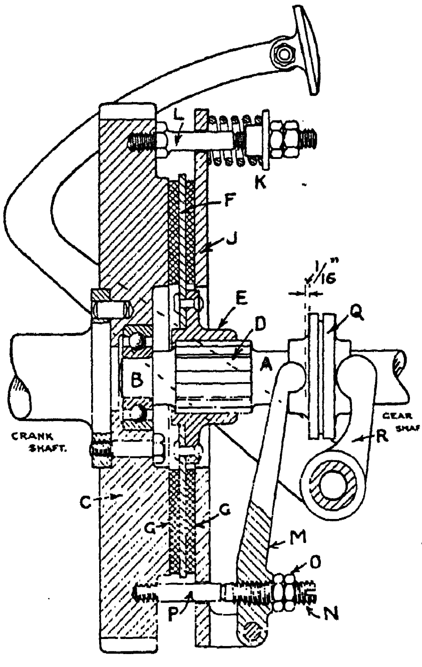


Fig. 112.—The Single Plate Clutch.

the compression of these springs before the presser plate can be moved to the right in order to free the friction plate from the engine drive.

The friction plate consists of a steel disc *F* having a flat ring of friction material *G* on each side of it. The plate *F* is riveted to a central boss *E* having internal splines, such that it can slide on the corresponding splines *D* on the gear shaft extension *A*. This shaft is provided with a spigot *B* having a ball-bearing mounted in the flywheel member.

Adjustments for the clutch spring compression are provided by means of threaded portions on the studs L, and adjusting and lock nuts. Similarly, the clearance between Q and the end of M (shown at its usual value of $\frac{1}{16}$ inch in the diagram) can be adjusted by means of the nuts O on the stud N.

The type of clutch shown in Fig. 112 is of the open pattern, the engine and gear box units being mounted separately on the chassis. The majority of modern cars, however, are fitted with a single engine, clutch and gearbox unit, the clutch driven member being carried through to the gear-box to form the primary driving shaft of the latter. This gives a more compact arrangement and obviates the necessity of universal joints between the engine and gear-box. A typical unit construction clutch and gear-box is shown in Fig. 115.

Fig. 113 depicts one of the clutch models made by Messrs.

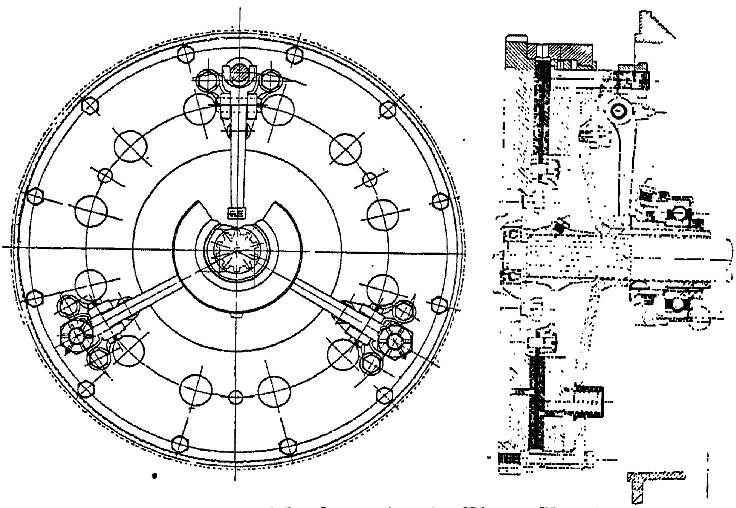


FIG. 113.—Meadows Single Plate Clutch.

Meadows, Ltd., and illustrates (on the left) the three symmetrically spaced operating levers for disengaging the clutch; one of these levers is shown in the right hand view.

The single plate clutch fitted to the Morris Eight car is shown in 'exploded' form in Fig. 114. This clutch has two

friction surfaces. The driving surfaces comprise two rings of bonded asbestos fabric, one attached to the flywheel cover-plate and the other attached to the pressure

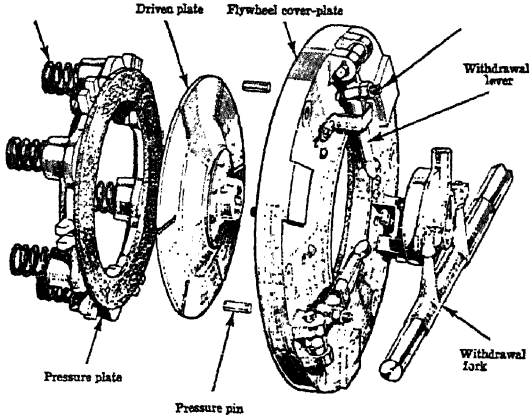


FIG. 114.—Morris Eight Single Plate Clutch.

plate. Six driving pins pass through the flywheel, presser or pressure plate and flywheel cover-plate, all of which therefore, revolve together.

The driven surfaces comprise both sides of a single steel disc splined to the driven shaft. Driving pressure for the clutch is derived from six helical springs housed between the pressure plate and the flywheel.

The clutch must be run dry, and persistent slipping of the clutch is usually an indication that oil has found its way into the clutch compartment, in which case it will be necessary to remove the drain plug in the bottom of the clutch housing and drain away any oil which may be present. Oil which may then be still adhering to the surface of the clutch plates will soon be burnt away after a little use.

If the clutch is allowed to slip continuously the centre driven plate very quickly becomes excessively hot, and the heat and friction will very soon destroy the surface of the fabric facings.

The assembled Morris Eight clutch is shown in Fig. 115; the end of the engine crankshaft, the flywheel and the splined clutch driven shaft, which is also the gear box primary drive shaft are also shown. The ball thrust and journal type bearing for the withdrawal lever collar is

seen immediately to the left of the operating lever whilst the clutch shaft spigot ball bearing is adjacent to the castle nut on the crankshaft end.

With this clutch a clearance of $\frac{3}{32}$ in. is arranged between the lever ends and the face of the withdrawal collar, when

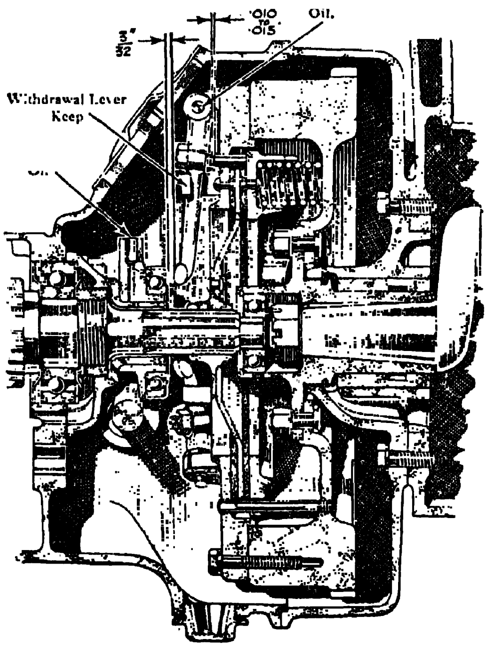


FIG. 115.—Morris Eight Clutch Unit.

the clutch is engaged; there is also a clearance of ·010 to ·015 in. between the end of the adjusting screw and the pressure pin when the lever is in contact with the restraining spring. The oiling points for the clutch withdrawal race and the lever pins are shown in the illustration; these items should be lubricated every 500 miles.

A sectional view of the Borg and Beck design of single-plate clutch that is used on several makes of British car is given in Fig. 116.

The clutch is of the single plate dry disc type, no adjustment for wear being provided in the clutch itself. An individual adjustment is provided for locating each lever

in manufacturing but the adjusting nut (14) is locked in place by means of a split pin.

A graphite release bearing (7), mounted in a cup attached to throw out fork is used, and a release plate (10) is

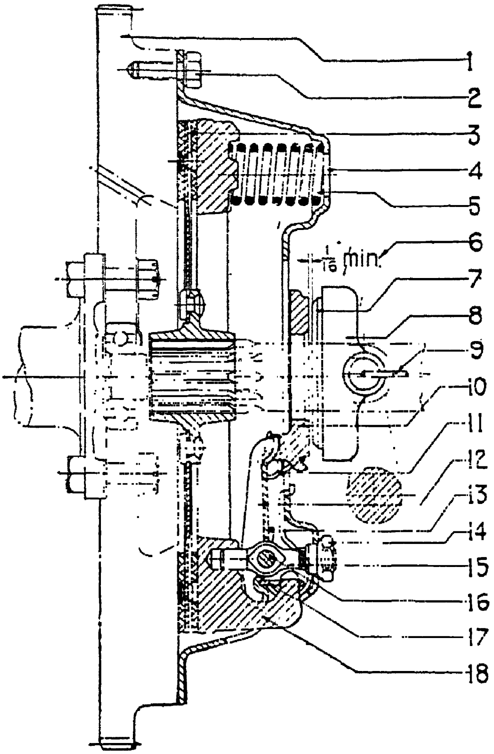


FIG. 116.—Borg and Beck Clutch.

attached to inner ends of release levers (12) by means of retainer springs (11). Release is accomplished by moving release bearing forward against the release plate (10). Each release lever (12) is pivoted on a floating pin (16) which remains stationary in the lever and rolls across a short flat portion of the enlarged hole in the eyebolts (15) (as shown in Fig 117). The outer ends of eyebolts extend through holes in the clutch cover and are fitted with adjusting nuts (14) by which each lever is located in correct position. The outer or shorter ends of the release levers engage the pressure plate lugs, by means of struts (17) which

provide knife edge contact between the outer ends of levers and pressure plate lugs, so eliminating friction at this point (see Fig. 117). Thus the pressure plate (18) is pulled away

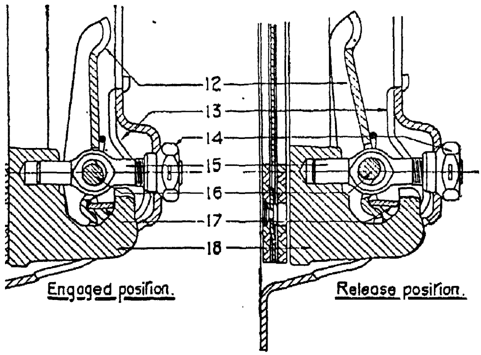


FIG. 117.—The Borg and Beck Clutch Release Levers.

from the driven plate (3) compressing the several small coil springs (5) which are assembled between the pressure plate and the clutch cover (4).

When the foot pressure is removed from clutch pedal the clutch springs force the pressure plate forward against the driven plate, gradually and smoothly applying the power of the engine to the rear wheels.

As the clutch facings wear the pressure plate moves closer to the flywheel face and the outer or shorter ends of the release levers follow. This causes the inner or longer ends of the levers to travel farther towards the gearbox, and decreases the clearance (6) between the release lever plate and the release bearing. The effect on the clutch pedal is to decrease the clearance or free travel under toe board. Adjustment is provided on all cars on the clutch pedal and its shaft to give the recommended clearance of $\frac{1}{16}$ in. (as shown at 6 in Fig. 116) between the graphite release bearing 7 and the release lever plate 10.

Later Improvements in Plate Clutches.—With the general adoption of the plate type of clutch certain improvements have been introduced as the result of experience in order to render the operation of the clutch as efficient as possible. The principal developments

have been in regard to its flexibility and long service with a minimum of maintenance attention.

Flexible Clutch Plates.—Instead of making the plate carrying the friction material from a solid disc of steel, it is now usual to split or perforate this plate in order

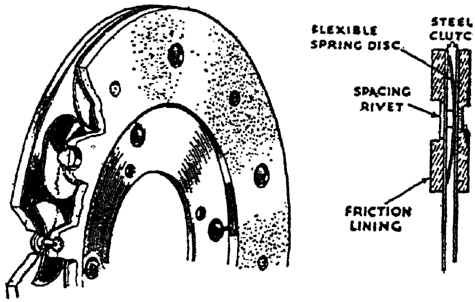


FIG. 118.—Flexible Clutch Plate Construction.

to give it a certain amount of end flexibility to ensure smoothness of engagement and more accurate alignment of the friction surfaces on their engaging members. Various methods of perforating the clutch plates are employed, a common one being to slit the driven plate at intervals, on its outer circumference. In some cases spring cup washers are arranged under the linings for the same purpose.

An example of a flexible clutch plate is the Newton and Bennett one depicted in Fig. 118. In this design the required flexibility is obtained by means of dished spring steel washers, each being fixed by a rivet to its plate member; the same rivet also holds the friction material to the plates. The dished washers are arranged with their curvatures facing in opposite directions, alternately. When the clutch is engaged these washers flatten out in the manner of a spring, thus giving a flexible action resulting in smooth engagement. The slots in the plates and the holes also, enable the air to circulate so as to cool the friction surfaces.

Another design of flexible clutch plate is the Don-Flex shown in Fig. 119. The clutch is of the inserted cork disc pattern and has double laminated spring steel plates held apart at the hub by a distance piece. The principal features

of this clutch plate are shown in the lettered indication lines of Fig. 119.

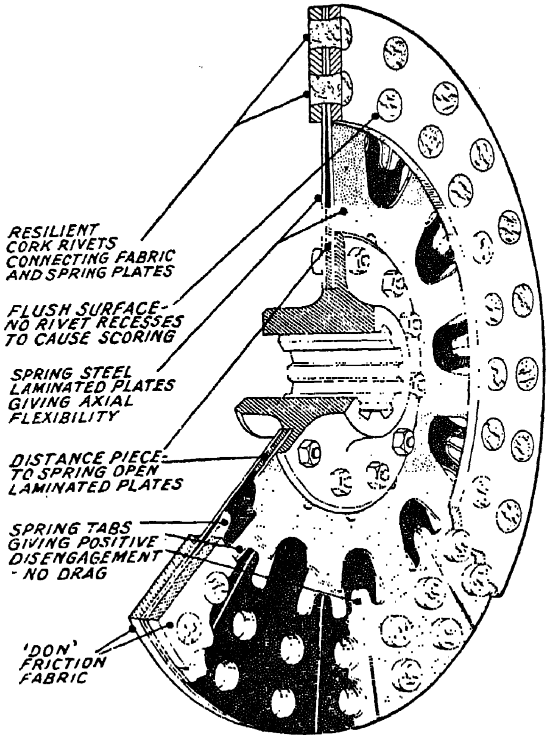


FIG. 119.—The 'Don-Flex' Flexible Clutch Plate.

Constant Pressure Clutches. An improved design of plate clutch is now available having a special resilient member and spring pressure system whereby instead of the clutch disengaging pressure increasing with the separation of the plates—or amount of depression of the pedal—it remains constant under all circumstances.

The advantages of this arrangement are that it permits of smoother engagement, compensates for the effects of wear and requires adjustment only after long periods of service. It has also good ventilation and therefore operates with minimum heating of the friction material.

Flexible Drive Device.—Instead of attaching the centra driven plate (having the friction material riveted to it

directly on to the boss that is mounted on the splined shaft, it has become the practice to interpose a spring drive, such that the engine power is gradually taken up when the clutch is engaged; the spring device also tends to absorb driving torque variations in the manner of a spring coupling between the engine and gear-box.

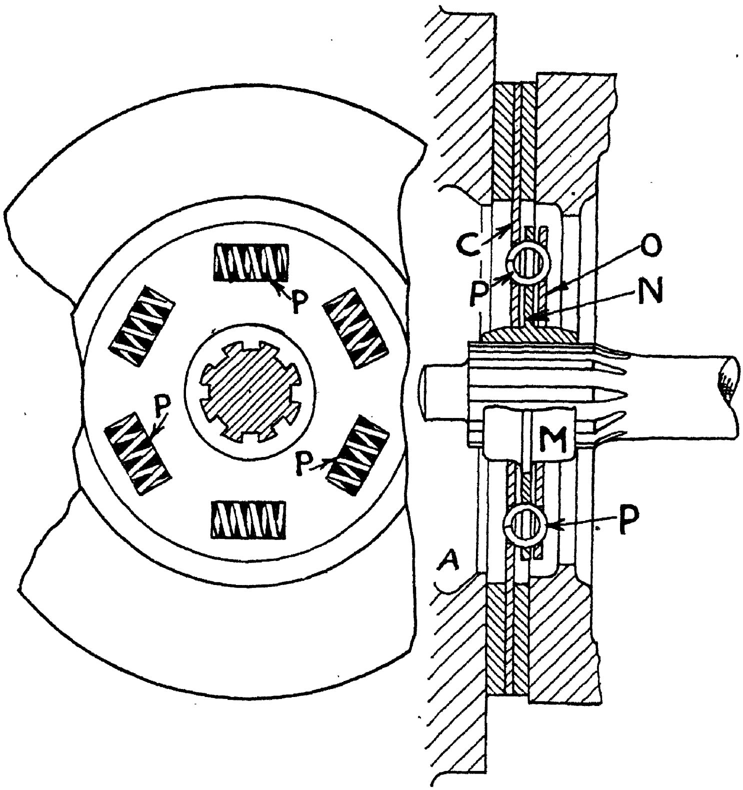


FIG. 120.—Flexible Drive Device.

Referring to Fig. 120 a sectional view of such a spring drive is shown on the right and an outside view on the left. In this method there is a central rigid plate *N* of small diameter formed on the splined sliding boss *M* which can slide on the gear box splined shaft. The pressure or central driven plate *C* is fitted round the boss but is not attached to it. Another thin plate *O* is

provided on the other side of the plate *N*. The driving connection between the plate *N* and the pressure plate *C* and its complementary plate *O* is by means of six compression springs *P* which fit into slots in all three plates. In this way the pressure plate *C* can only transmit the drive to the plate *N* through the springs when the clutch is engaged, thus giving an elastic coupling between the two members.

The Borg and Beck compound cross cushion driven

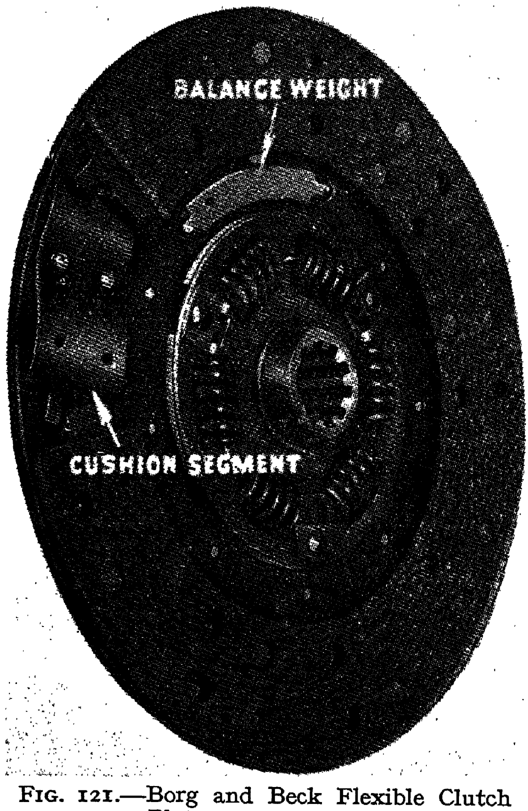


FIG. 121.—Borg and Beck Flexible Clutch Plate.

clutch plate shown in Fig. 121 employs the cushion springs previously described and in addition has a flexible pattern driven steel plate provided with peripheral slits and a series of perforations with slightly bent, flexible fingers

to give a smooth engagement between the driving and driven plates. The splined shaft upon which the driven plate slides is of 3 per cent. nickel chromium case-hardening steel.

Referring to the component parts of the Morris Twelve Four clutch shown in Fig. 122, the clutch pressure plate

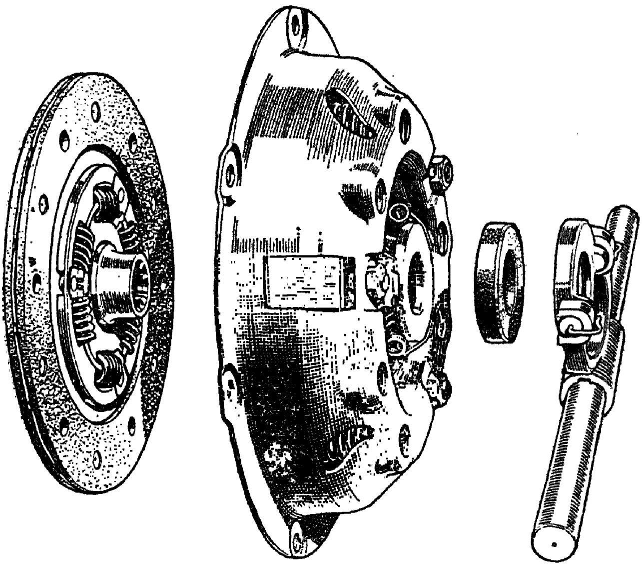


FIG. 122.—Components of Morris Twelve Clutch.

on the left shows the coil springs mentioned. It also shows the flexible plate, the edges of which are divided at intervals around the periphery of the plate.

Graphite Thrust Bearing.—In many of the later model clutches the usual ball-thrust bearing of the withdrawal or 'throw-out' member has been replaced by a block (of annular form) of solid graphite. The advantage of this material is that it will withstand the thrust with a low frictional co-efficient and, being self-lubricating needs no lubrication attention; moreover, if used correctly it has a very long period of useful service. It is important, however, to limit the travel of this thrust member, otherwise it may become damaged; a special adjustment is often provided for this purpose. Referring to Fig. 122 the

graphite thrust washer is shown between the withdrawal shaft on the right and the clutch casing.

Cork Insert Clutches.—In this type of plate clutch instead of using a ring of friction material a large number of cork studs, or inserts is employed for the two friction surfaces of the driven plate; this type of clutch has been used with satisfaction on certain Morris and Hudson cars.

The use of cork has certain advantages, for this material is more resilient than friction fabric materials so that there is a smoother take-up of the engine drive; moreover, the cork gives a very long period of service without

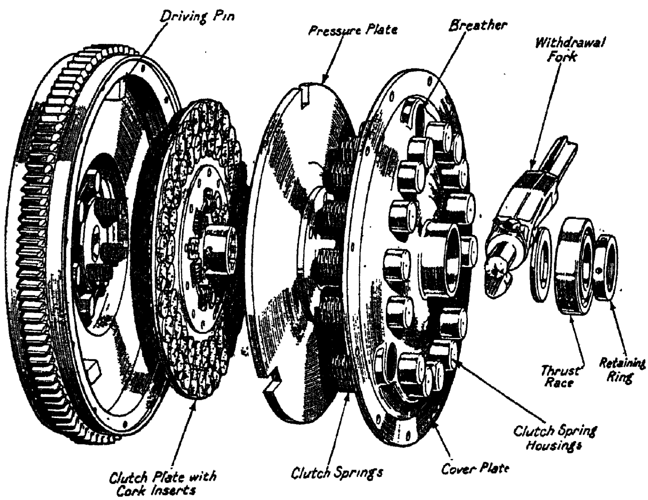


FIG. 123.—Typical Cork Insert Clutch.

attention, other than that of lubrication—since this type of clutch must run in oil. Usually the requisite quantity of oil is fed from the crankshaft rear main bearing, but in certain other instances the clutch runs in a sealed casing in a mixture of equal parts of engine oil and paraffin.

Fig. 123 shows the Morris Series III Twenty-Five car clutch in the dismantled state, each of the components being clearly indicated. It should be mentioned that, owing to the yield of the cork inserts under the clutch spring pressure the pressure plate takes up a position closer to the flywheel, there being a correspondingly

greater movement of the clutch pedal after engagement, than with the solid friction material type of clutch.

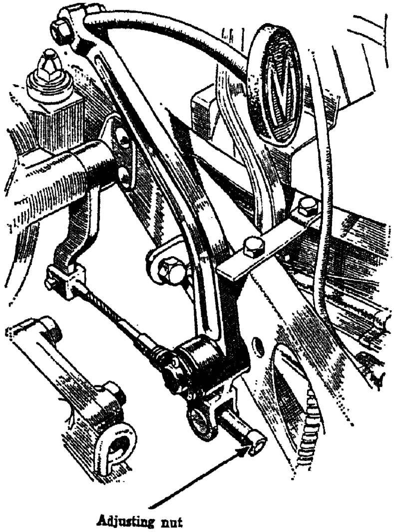


Fig. 124.—Clutch Pedal Adjustment.

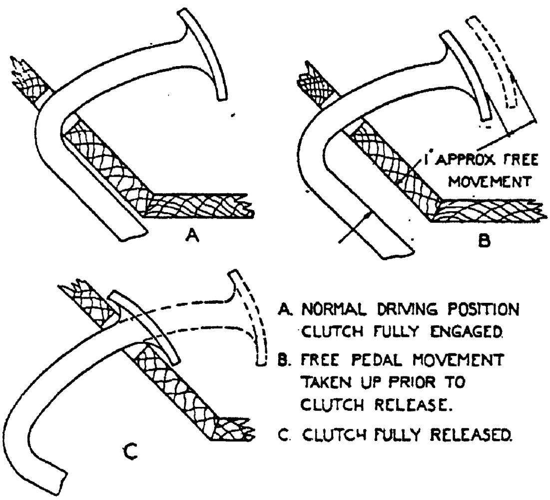


Fig. 125.—Illustrating Method of Adjusting Clutch Pedal Position.

Clutch Pedal Adjustment.—In all cases there is an adjustable member in the clutch operating mechanism

to enable the clutch pedal position to be adjusted to the bedding-down of the friction material and to give correct disengagement (and engagement movement) the pedal touching the foot-board. Fig. 124 Morris Twenty-Five cable adjustment for t pedal.

Whatever the method of adjustment that is on the chassis, the pedal should be moved away footboard. In the case of the Borg

Plate Clutch Calculations.—Calculations con clutches involve a knowledge of the engine t is obtained from the well-known relation :—

$$=\frac{T.N.}{5252}$$

inner radii of the friction

$_1{}^3-R_2{}^3)$ lb. ft. per single friction surface

ch per sq. in. and
of frictional force at
the mean radius

$$\text{Then } F=\frac{T\times 1}{Rm}$$

$$= \frac{F}{\text{Area of Single Disc}} = \frac{F}{\pi\,(R_1^2 - R_2^2)}$$

$$= \frac{12T}{\pi\, Rm\, (R_1^2 - R_2^2)}$$

The number of steel plate members will be $(n+1)$. If P is the total spring pressure on the friction surfaces,

The value of p, for bonded asbestos on cast iron should not exceed 45 lbs. per sq. in.

Another formula for clutch torque capacity is as follows :

$$T = N \times f \times p \times k$$

where T = torque in lbs. ft.

N = number of friction surfaces.

P = total effective pressure on linings.

k = radius of gyration of linings.

f = coefficient of friction of linings (usually taken as 0·25 average slipping and 0·30 average break-away).

The radius of gyration k in feet for a flat circular ring about a perpendicular central axis is :—

12

where R = outside radius of ring.

r = inside radius of ring.

The Clutch Spigot.—An important feature of most clutches is the support bearing for the engine side of the clutch shaft. This is usually in the form of a ball-bearing or one of the 'oil-less' bush patterns. In earlier designs of clutch, plain bushes were frequently employed, but these have been discarded in favour of the other types mentioned. The lubrication of the spigot bearing is an important matter, since any wear in this bearing will give rise to clutch 'wobble' and possible noisy operation. Special provision should be made for lubricating this bearing periodically, unless it is of the automatically lubricated type.

Care of Single Plate Clutches.—These clutches seldom

give trouble if used properly, *i.e.*, engaged gently every time.

If the clutch is inclined to slip under load, the compression springs should be tightened a little. After long periods of use, or in cases of excessive slipping, it is advisable to dismantle the clutch and to wash all the parts in petrol or paraffin. As most single plate clutches run 'dry' it is important to keep oil and grease away from the friction surfaces.

Multiple Plate Clutches.—In the case of *Multiple Plate* or *Disc* clutches, a number of metal and friction material discs are used; this type of clutch is employed for larger horse-power transmission; the use of several discs also reduces the operating pressure required, and the overall diameter of the clutch.

The Double Plate Clutch.—In this case there are two plates having friction material on each side, and three other members between which they work, viz., the rear face of the flywheel, the floating steel disc between the two friction plates and the forward face of the clutch pressure plate; normally, all the surfaces are pressed together with springs. The friction plates are covered partially with friction fabric or with cork inserts.

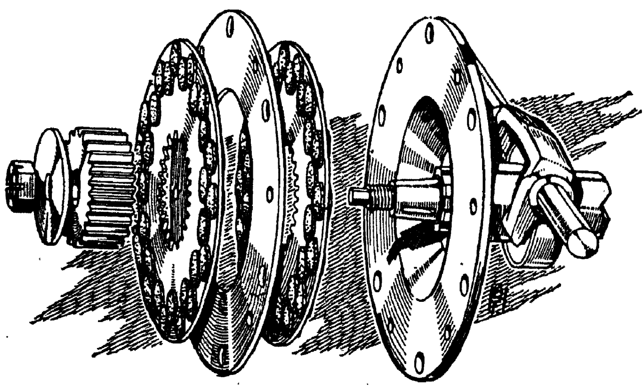


FIG. 126.—A Double Plate Cork Insert Type Clutch.

Fig. 126 illustrates a double plate clutch, which uses cork inserts as the friction material; the plates run in oil and work very smoothly when the clutch is engaged

or disengaged. Six clutch driving pins pass through the flywheel, the floating plate and the pressure plate, all of which consequently revolve with the engine.

The driven surfaces comprise a double line of cork

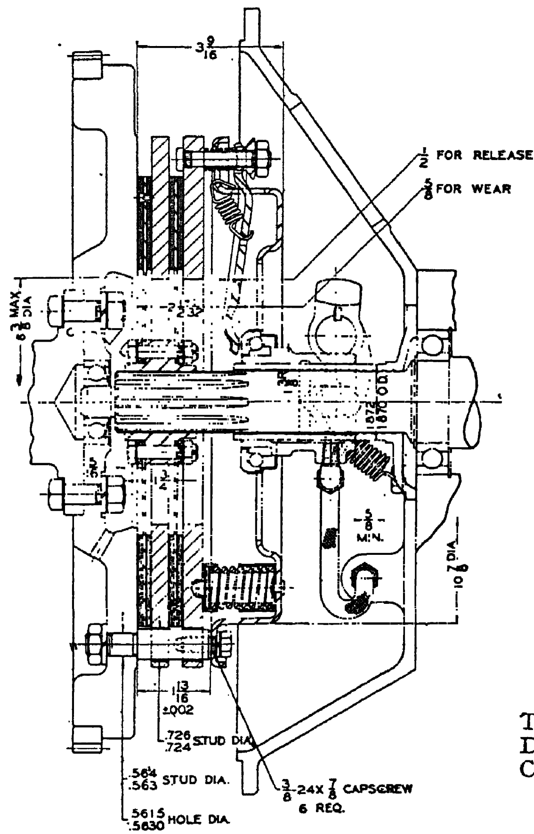


FIG. 127.
The 'Long' Heavy Duty Double Plate Clutch.

insets in two steel plates, the plates themselves being mounted on a toothed driving hub keyed on to the driving shaft carrying the direct drive, i.e., top speed, one plate being between the flywheel and the floating plate, the other between the floating plate and pressure plate.

Driving pressure for the clutch is derived from helical springs on the driving pins outside the pressure plate. A clutch stop, of the adjustable type, is fitted to prevent the

clutch spinning after the pedal has been depressed; is useful when gear-changing.

A typical double plate dry clutch as used for nigh powered cars and certain commercial vehicles is shown in Fig. 127. This is known as the Long pattern and the particular model illustrated will deal with a transmitted torque of 550 lbs. ft. It contains 18 pressure springs and six clutch release levers each with a leverage ratio of 5 : 1. The clutch shaft front spigot runs in a heavy duty ball bearing; a ball thrust release bearing is provided for operating the clutch. The complete clutch weighs about 100 lbs.

Fig. 128 illustrates an earlier pattern multi-disc dry

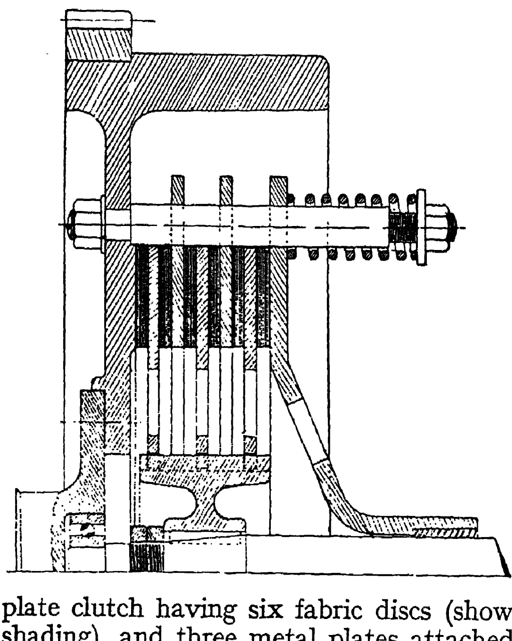


FIG. 128.

Simple Multiple Plate Clutch with Ferodo-faced Discs.

plate clutch having six fabric discs (shown by the heavy shading), and three metal plates attached slidably to the central boss shown keyed to the right-hand central gear drive shaft. The outer and inner flywheel surfaces, and the two annular metal discs driven by the flywheel can also be seen in Fig. 128. It will be observed that the necessary engagement pressure is obtained by a series of small coil springs, guided by the studs attached to the flywheel; these studs pass through the two metal discs mentioned,

and transmit the engine drive to them; these discs can slide a little on the studs. The clutch is disengaged by moving the extreme right-hand engine driven member to the right; this action releases the pressure of the springs on the metal and fabric discs. This type of clutch works very smoothly, and without slip or chatter; it wears well for extremely long periods. The clutch spring's pressure can be adjusted by means of the nuts on the guide studs—one of which is shown in Fig. 128.

Another type of multiple plate clutch employs a number of metal plates, half of which are engine driven, and half attached to the gear-drive shaft. These plates run in thin oil. In the previously adopted Hele-Shaw clutch

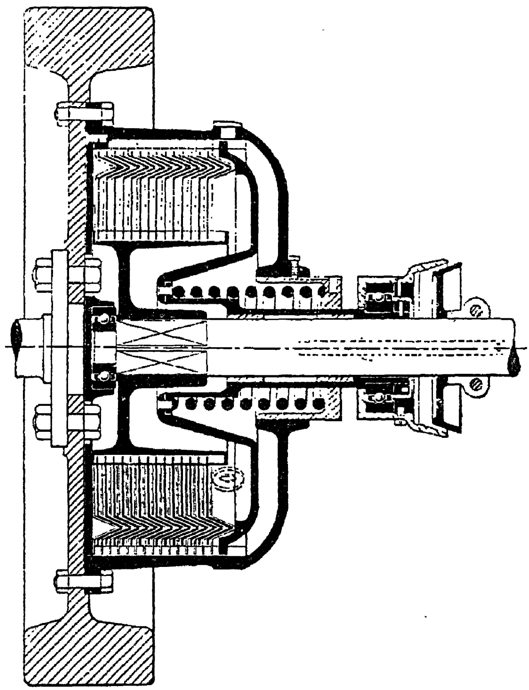


FIG. 129.

The Hele-Shaw Multiple Metal Plate Clutch.

(Fig. 129), ten such plates were used for each purpose, the plates being indented conically over part of their surface, in order to give a V-shaped engagement surface. Holes were provided in the plates for lubrication purposes.

It was very essential to keep the plates very clean, and lubricated only with the proper oil.

Reduced Pressure Clutch Spring.—The ordinary clutch spring for keeping the plates of the clutch firmly together when transmitting the drive, is the helical wire compression type. With this form of spring the process of declutching requires a greater pressure than that required to hold the plates together; this means that the driver must exert a heavy pressure on the clutch pedal to hold the clutch out of engagement; further, the pressure increases with the amount of pedal depression.

An entirely new design of clutch spring, shown in Fig. 130 and known as the Ingersoll, enables the clutch release

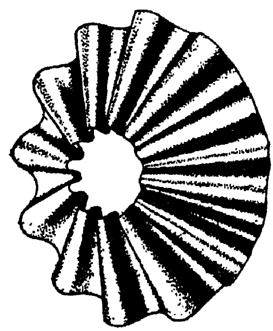


FIG. 130.
The Ingersoll Reduced Pressure Spring.

pressure to diminish as the clutch pedal is depressed; this form of spring is used on certain American cars, including the Buick one.

As shown in Fig. 131, when pressure is first applied to this spring in the direction of its axis, the resistance of the spring to deflection increases substantially in proportion to the deflection, but the rate of the spring gradually decreases, and with a deflection of $\frac{3}{4}$-in. the pressure attains its maximum value of 300-lbs. With the clutch engaged, the deflection is only slightly greater, and the pressure exerted by the spring is still close to the maximum value of 300-lbs. If the spring is deflected further, as when disengaging the clutch, its pressure decreases, and for the deflection corresponding to full disengagement it amounts to only 225-lbs. With a straight-line characteristic, a spring which shows 300-lbs. pressure for a deflection

0.95-in. would show 500-lbs. at 1.6-in. deflection. This characteristic is obtained from what may be described as a spring of conical-disc type with deep radial folds or cor-

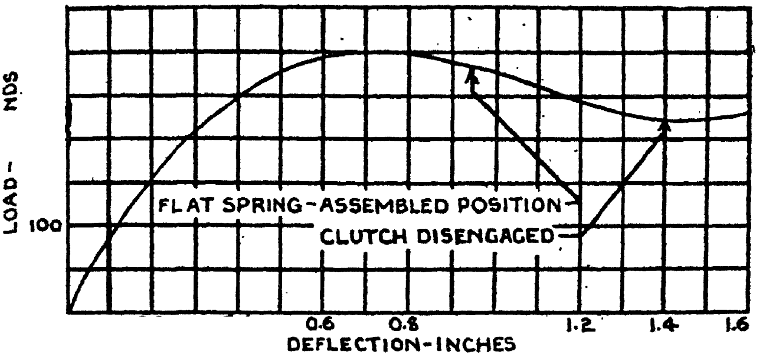


Fig. 131.—Load-deflection Curve for Ingersoll Spring.

rugations. The pressure of the spring increases as it approaches the flat position, and decreases as this position is passed.

The Semi-Centrifugal Clutch.—The types of clutches previously described are all designed with pressure springs of sufficient strength to prevent any slipping of the friction members when the maximum engine torque is being transmitted through the clutch. In the higher-powered engines which may develop from 50 to 150 b.h.p., the clutch spring pressures may be considerable, thus requiring appreciable physical effort to disengage the clutches; this effort may become fatiguing to the driver. A new type of clutch was evolved to overcome this difficulty, by employing springs of relatively low compression to enable the friction surfaces to take only the ordinary engine torques. In order to take the maximum engine torque effects a centrifugal device is used, such that as thc engine speed is increased the centrifugal force also increases (but at a higher rate); this centrifugal effect is utilized to increase the pressure between the friction surfaces. In this way the driver requires a relatively light effort to disengage and hold out the clutch at low speeds, slipping at the higher speeds being prevented by the centrifugal loading of the clutch plates.

Referring to Fig. 132, which shows a Borg-Warner in section there is a series of three hinged and weighted levers arranged at equal intervals; one of these is shown in the right-hand view to an enlarged scale. The fulcrum, or hinge, of this lever is at *D* (left-hand illustration) and it is hinged to the pressure plate just above this; needle bearings are used for this purpose. The upper end *A* of the lever is weighted. It will be understood that at low

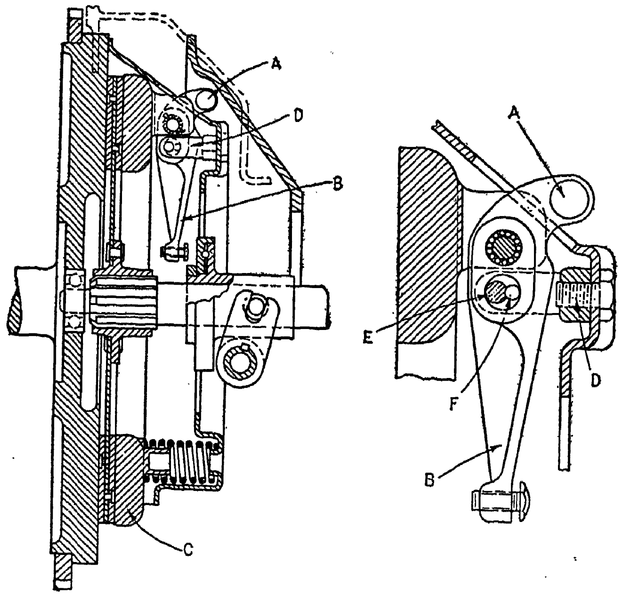


FIG. 132.—Semi-Centrifugal Clutch.

engine speeds the ordinary light compression springs (one of which is shown near *C*) prevent clutch slip, the centrifugal effect upon the weighted portion *A* being very small.

As the engine is accelerated the end *A* tends to move outwards under centrifugal action, thus causing a torque about the centre D, which results in a force normal to the pressure plate at the needle bearings; the effect of this is to increase the pressure on the pressure plate. The centrifugal force increases as the square of the speed, so that the pressure preventing slip is adequate at full engine load or torque conditions.

Instead of using a plain pin bearing at *D*, this particular clutch has flat-sided pins *E* carried by the posts *D* (right hand diagram). Between the flats of the pins and the edge of the hole are rollers *F*. The object of this type

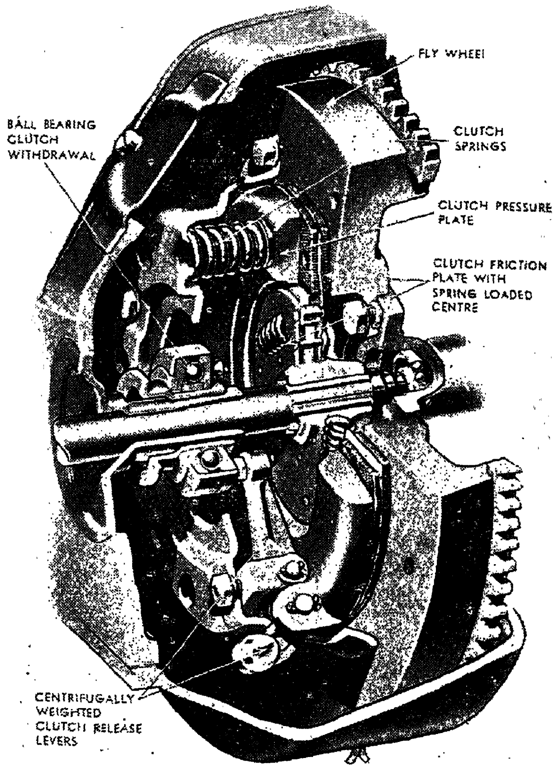


FIG. 133.—The Vauxhall Semi-Centrifugal Clutch.

of bearing is to give a straight line movement for the pressure plate bearings.

The Vauxhall semi-centrifugal clutch shown pictorially in Fig. 133 will enable the reader to follow out the various members previously referred to.

The Fully Centrifugal Clutch.—From the semi-centrifugal type it is not difficult to understand that, by making the centrifugal device to operate positively from the

lowest engine speeds—or idling speeds—it is possible to make the clutch entirely automatic.

In this case the light compression springs between the clutch casing and the pressure plate *C* (Fig. 132) are dispensed with, but in their place are fitted other compression springs between the friction plate and pressure plate to hold these apart when the engine is at rest or is merely idling at speeds up to about 400 to 500 r.p.m. The engine is thus entirely uncoupled from the gearbox.

If now the engine is accelerated, the centrifugal action on the pivoted levers increases and, overcoming the spring pressure, brings the pressure and friction plates together with increasing pressure. At first there is a certain amount of slip, but beyond about 1,000 r.p.m. the centrifugal effect is sufficient to overcome this and the full engine torque can be transmitted without any further slip.

When the engine speed diminishes to about 600 r.p.m. the clutch is automatically disengaged and the engine is thereby disconnected from the gearbox.

It will be apparent from this, that no clutch pedal is

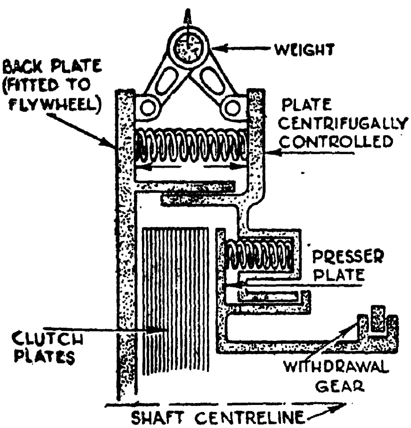


FIG. 134.
Principle of The Newton Clutch.

necessary, the only driving control being the accelerator pedal.

Typical examples of fully centrifugal clutches are the Newton and Talbot ones.

The principle of the Newton centrifugal clutch, that has been used on Riley and other cars is shown in Fig. 134.

When the engine is idling the compression springs hold the clutch plates in the free position, but as the engine is accelerated the weighted levers tend to move outwards thus drawing the right hand plate and with it the presser plate so as to force the clutch plates together. It will be observed that there is a positive withdrawal gear that can be operated when desired.

The Talbot Traffic Clutch.—This is a form of centrifugal clutch combined with a free-wheel device to prevent the

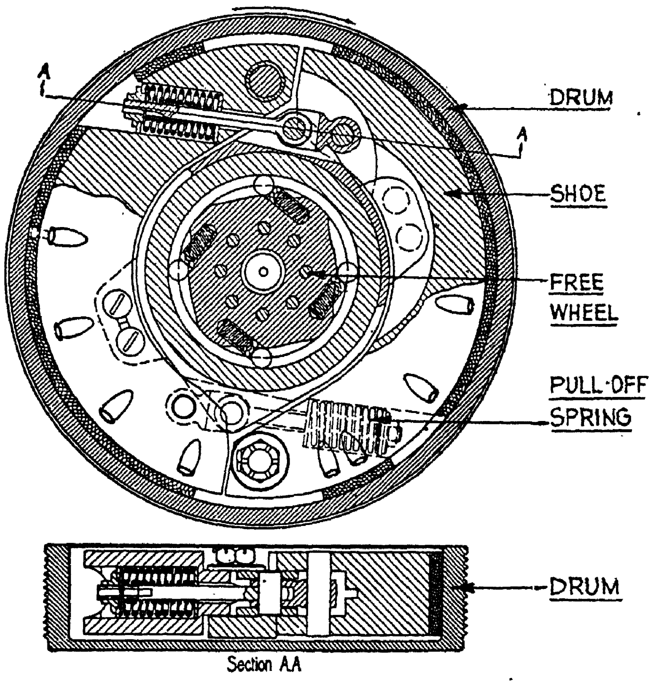


FIG. 135.—Talbot Traffic Clutch.

car from coasting should the clutch be disengaged when the car is descending hills.

The Talbot clutch (Fig. 135) consists of a kind of brake drum formed on the engine end of the main gear shaft, and a pair of heavy brake shoes hinged on to a flanged member attached to the engine crankshaft. Normally, the brake shoes are held together by means of springs so

that when the engine is not operating or is idling only, there is no connection between them and the brake drum on gear shaft. When the engine is accelerated, however, the brake shoes move into engagement with the drum and thereby transmit the drive through to the gearbox. Should the engine speed fall below about 900 r.p.m., the brake shoes commence to lose their 'centrifugal' grip on the drum and slipping occurs; when the speed drops to 400/500 r.p.m. the shoes are entirely out of contact with the drum so that the engine can 'idle' without driving the gears in the gearbox.

It will be seen that this clutch operates in a similar manner to the fully centrifugal plate clutch previously described.

In regard to the free-wheel device, this is arranged inside the brake shoe members and is of the spring-controlled rollers and inclined plane type; the inner part of the free-wheel is the actual driving member. When the car is over-running the engine—as when descending a hill with the engine running below about 900 r.p.m.—the drive from the road wheels is transmitted through the gearbox to the engine so that no free-wheeling occurs under the circumstances; if this device was not fitted the engine and transmission would, of course, be disconnected when the car was coasting with engine running slowly.

The speed at which the clutch shoes come into operation can be regulated by means of an adjustable nut on the brake shoe mechanism.

The Daimler Fluid Flywheel.—This interesting device replaces the clutch of the ordinary motor-car, and substitutes in its place a driving member on the engine-shaft and a driven member on the gearbox main shaft. The two members are not in contact in any way, but are separated by a fluid of suitable viscosity; this fluid transmits the drive from the engine to the gearbox member.

The driving member is integral with the outer casing of the flywheel, which is bolted to the engine-shaft. Fig. 136 shows the series of cup-shaped pockets separated by radial webs which are formed upon the inner surface of this member. A small gap separates the driving member from the driven member so that the latter may be free to rotate.

The driven member is also shown in Fig. 136, and has similar pockets and webs formed in the surface of this member, which is free to rotate within the flywheel casing, but is keyed to the gearbox driving shaft. The driven shaft runs upon a bearing within the flywheel casing;

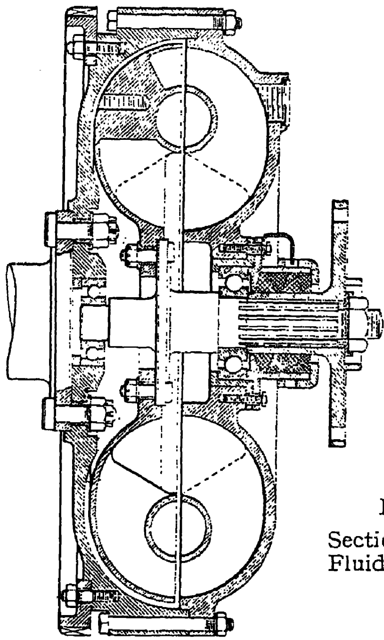


FIG. 136.
Sectional View of Fluid Flywheel.

there is in addition an oil retaining ring and spring on the inner side of the bearing to prevent leakage.

The fluid flywheel transmits power in the following manner: Assuming the car be stationary and the engine started, the rotation of the driving member by the engine causes the oil in its cells to flow towards their outside periphery. From here, as the driven member is yet stationary, the oil flows past the outside periphery of its cells, through them, and past their inside periphery to the inside periphery of the driving member cells, and from there back again to their outside periphery. In other words, the oil starts on a circulatory motion between the cells of the driving and driven members.

In passing from the webs of the driving to those of the

driven member, the oil is retarded in velocity and, therefore, imparts some of its momentum to the driven member, thereby setting the latter in motion. Since, even when the driven member has attained full speed, the load on it causes

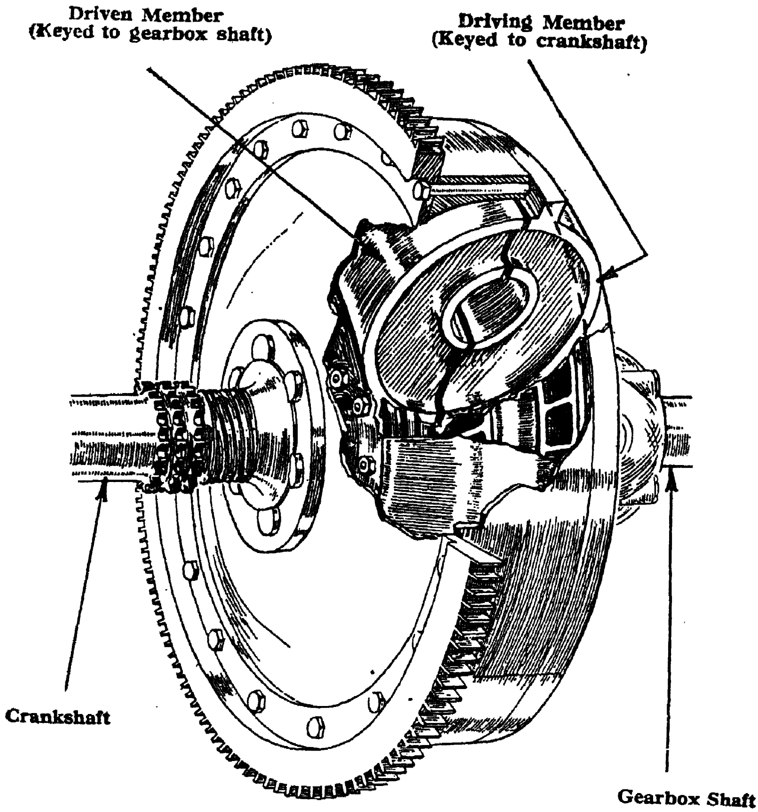


FIG. 137.—The Daimler Fluid Flywheel.

it to lag behind the driving member, the centrifugal forces in the latter are always larger than those in the driven member, so that the circulatory motion of the oil and, therefore, the transmission of power from one to the other, is always kept up.

At ordinary speeds the oil needs but little retardation to develop the required driving torque, hence the lag or

'slip' between the driving and driven members is insignificant. At low engine speeds, however, the 'slip' can become 100% at full torque, thus providing the condition that the engine can develop full torque in gear without moving the car. This occurs at about 600 r.p.m.

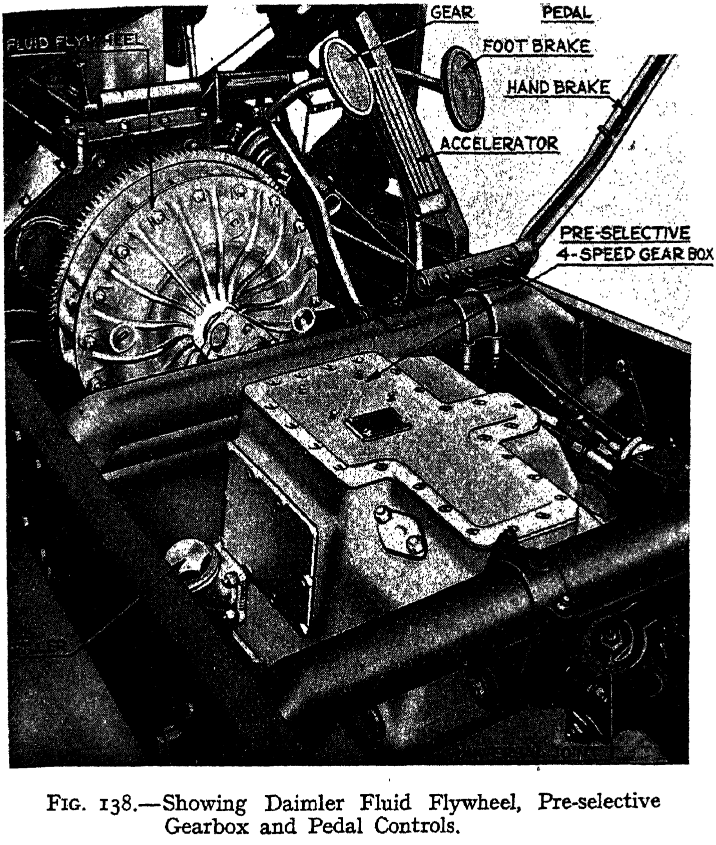


FIG. 138.—Showing Daimler Fluid Flywheel, Pre-selective Gearbox and Pedal Controls.

The curve, Fig. 139, shows the lag or 'slip' between the driving and driven members of the fluid flywheel at various speeds. The full line is the 'slip' under full throttle conditions, while the dotted line shows the normal amount of 'slip' existing when the car is travelling at a steady speed on a level road. The important point to note is that

while the 'slip' at very low engine speeds (on full throttle) can be 100%—as it is when starting off—it falls off exceedingly rapidly but smoothly with increasing speed. As soon as the car is travelling at a normal speed the 'slip' is of the order of 2% only.

In regard to the maintenance of the fluid flywheel, this is merely a matter of keeping the casing filled to the level

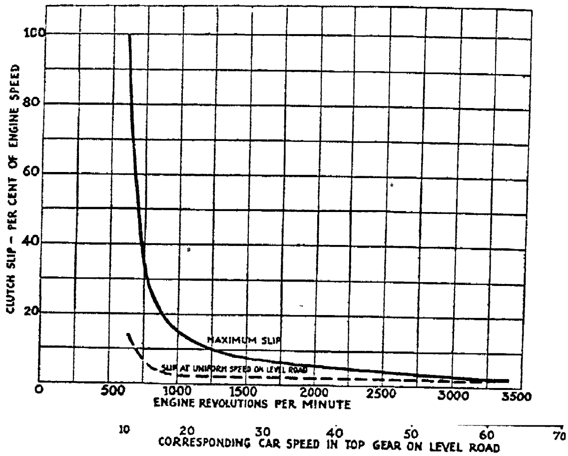


FIG. 139.—Percentage Slip of Daimler Fluid Flywheel at Different Speeds.

of a special filler-hole, when the flywheel is turned to the position recommended by the makers (the boss is usually about 4 inches off centre). After filling with Daimler engine-oil the plug should be replaced tightly. Any plug removed from the flywheel should be replaced in the same hole as otherwise the balance of the flywheel will be affected.

The Daimler fluid flywheel is used on Daimler cars in conjunction with the Wilson pre-selective gearbox described elsewhere in this volume.

The Vulcan-Sinclair Hydraulic Clutch. This type of hydraulic coupling is somewhat similar in principle to the one previously described. It has a wide application including; not only automobiles but also marine and stationary industrial plant, rail cars, winches and cranes, etc.

The principle of the coupling is illustrated in Fig. 140, in which the impeller member A is the input, or engine

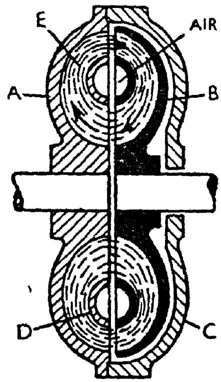


FIG. 140.
Principle of the Vulcan-Sinclair Coupling.

drive member, whilst the runner B is the output or gear-box drive member. A cover C is attached to the impeller. An annular core ring of semi-circular section, shown at D, is formed with one half in the impeller and the other half in the runner.

With the unit filled with oil and the impeller rotating the liquid in the passages between the vanes of the impeller flows radially outwards under centrifugal force, as indicated by the arrows at E. It passes across the gaps separating the impeller from the runner and flows radially inwards between the vanes of the runner until it reaches the inner diameter of the working circuit where it returns across the gap into the inlet of the impeller and the cycle is repeated; it thus gives a rotating vortex ring of liquid having a definite kinetic energy depending upon its mass and velocity. Whereas the transmission of power from the impeller to the runner is dependent upon the existence of this vortex movement, the actual torque is created by the loss of rotational energy and the release of the kinetic energy as the liquid is forced radially inwards between the vanes of the runner.

For automobile purposes the tank and pump are dispensed with and a simpler design is made possible. Fig. 141 illustrates a Vulcan coupling as used for automobiles. It contains its own reservoir chamber on the back of the

runner or driven part. The reservoir chamber A is connected with the working circuit by a number of tubes *B* extending into the centre of the vortex ring *C* to permit of interchange of liquid from the working circuit *D* to the reservoir, and vice versa.

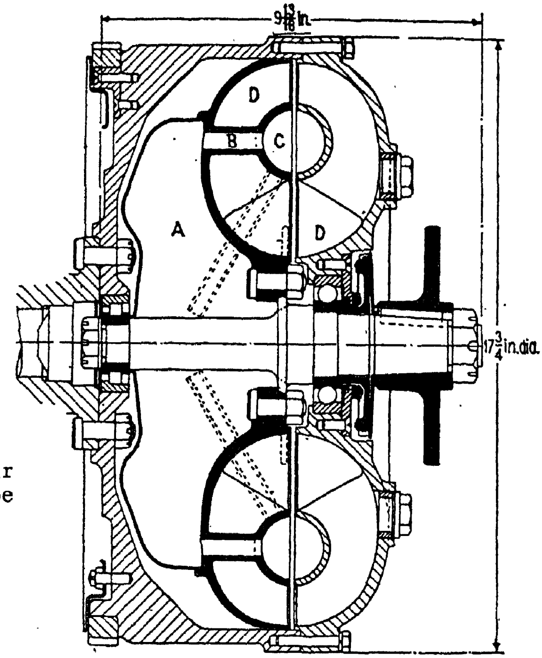


FIG. 141.
Vulcan-Sinclair Automobile Type Coupling.

The objects of the reservoir chamber may be enumerated as follows. (1) To remove part of the oil from the working circuit under starting conditions, i.e., between 100 and 50 per cent. slip, so as to reduce the drag torque or creeping tendency and to assist the engine in picking up the load more readily when the throttle is opened. (2) To return the oil into the working circuit automatically when a certain speed is reached, so as to reduce the slip to a very low value at normal road speeds. (3) To act as an expansion chamber so that the pressure rise in the coupling is of a negligible order, even if it is stalled at high torque, until the temperatures are in excess of any met with in ordinary usage. (4) To separate air from the oil so that the working circuit is kept centrifugally charged with air-

free oil whilst the air collects near the centre of the reservoir chamber.

The traction coupling thus permits the use of a rather larger diameter of working circuit than is possible with a

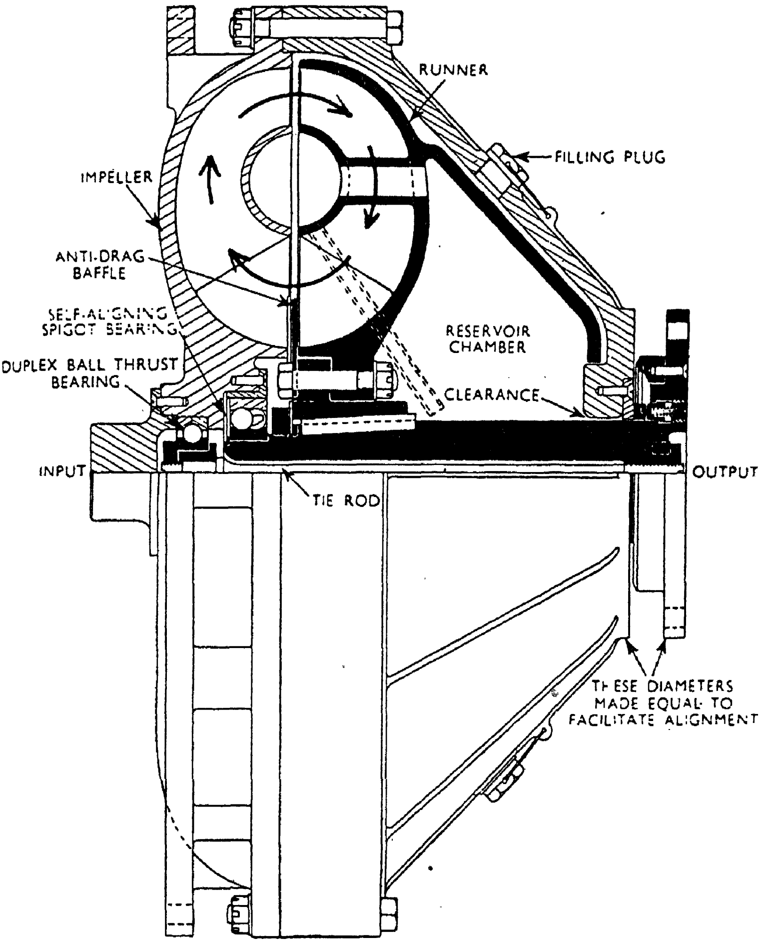


FIG. 142.—Improved Vulcan-Sinclair Coupling with Low Idling Drag.

fluid flywheel, and since the working circuit is partly emptied when starting, the traction coupling then has a lower drag torque; also it has less slip at normal running speeds than the ordinary fluid flywheel.

The Vulcan-Sinclair coupling shown in Fig. 141 has suitable dimensions for transmitting about 130 H.P. at 2,400 R.P.M.

In order to reduce still further the torque (residual) when the engine is idling, the coupling design shown in Fig. 142 is often employed. When the runner is stopped the working circuit partially empties into the reservoir, thus reducing the torque and, further, the anti-drag baffle interferes with the circulatory flow of the remaining liquid; in this way it is possible to *reduce the value of the drag torque to about one-quarter* of that of the usual fluid flywheel, when the engine is idling.

This type of coupling is of special interest in association with the Cotal epicyclic gear box, using electromagnetic clutches, as it provides a flexible drive and 'cushions' the variation of torque at the moment of changing gear.

The Vulcan-Sinclair coupling, which is a power transmitter and not a torque converter, has been used upon Singer cars with satisfactory results.

Free-Wheel Clutches.—A similar principle to that of the ordinary pedal bicycle free-wheel has been applied to motor vehicles to enable them to take advantage of any declines, or the momentum of the vehicles themselves, for free-running purposes.

Several types of motor-car free-wheel devices, including the Humphrey Sandberg, Millam and De Lavaud mechanisms, have been fitted to European cars.

The free-wheel device enables the car to free-wheel, as it were, on any down-gradients, or when the accelerator is released. At the same time it does not prevent the car from being driven positively in a forward direction. In most designs there is an automatic or hand-operated device to 'lock' the free-wheel when required, as when the car is reversed or when it is desired to use the engine as a brake down long, steep hills.

The advantages of the automobile free-wheel device are that it saves the transmission from wear whenever the car free-wheels, since it disconnects the engine and gearbox from the propeller shaft, leaving the rest of the transmission under light loads only. Since the engine and gearbox are disconnected when free-wheeling, there is less frictional resistance to the movement of the car when coasting, and,

further, as the car actually coasts on most downward slopes, an appreciable amount of petrol is saved, viz., from 15 to 20 per cent.

The principal advantage of the free-wheel device, however, is that it makes gear-changing up or down much easier at speeds below about 20 m.p.h.

As there is no engine and gearbox friction when free-wheeling the brakes must be used more forcibly to pull the car up when stopping.

The usual place for the free-wheel device is at the back of the gearbox, between the latter and the front end of the propeller shaft. The Humphrey Sandberg device used inclined rollers which were in contact with curved conical surfaces of the driving and driven members. These surfaces had special mathematically calculated shapes, so that they had the following special property, namely, if an outer hollow conoidal surface be arranged concentric with the inner conoidal member, a cylindrical roller placed at the right inclination to the axis (this is usually about 24°) will make line contact with both the surfaces if the conoids are of the right dimensions. When the inner member, say, was rotated in a certain direction, the rollers simply free-wheel, that is, they rotated just as in a roller bearing. If,

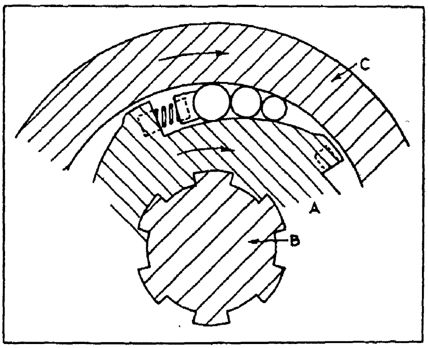


FIG. 143.—Free-wheel Principle.

on the other hand, the inner member was rotated in the reverse direction, the rollers tended to ride up and jam, thus giving a positive lock against rotation in that direction. With this arrangement any wear which may occur is distributed uniformly over the whole of the surfaces; the

inner cone then moves a little closer to the outer one, so that the wear has no practical effect on the working of the device.

It was possible to arrange for the movement of one or other of the cone members on its axis so that the free-wheel action can be prevented, and *a solid lock obtained.*

The principle of one design of free-wheel used on later cars is illustrated in Fig. 143. It consists of a central driving member *A* mounted securely on the castellated shaft *B* which forms the rear end of the gearbox driven main shaft. The outer annular member *C* is attached to a member connected through the usual front end universal coupling to the propeller shaft.

Between the members *A* and *C*, in each of the wedge-shaped spaces—of which there are usually three or four in the complete free-wheel—is a series of three hardened steel rollers of different sizes. In the positive drive position shown by the arrows these rollers are forced

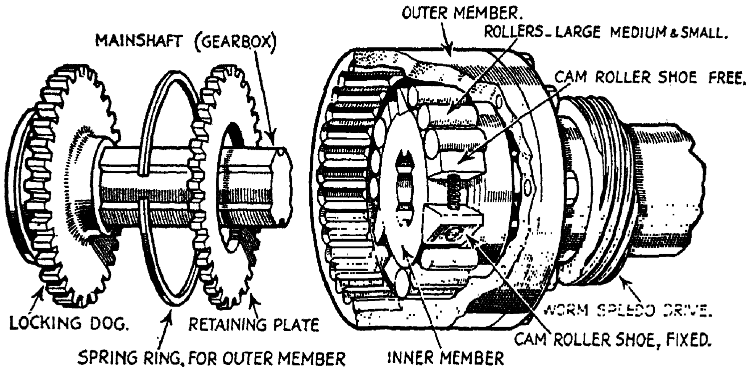


FIG. 144.—The Rover Free-wheel Unit.

to the right by the spring pad and become wedged between *A* and *C* as a result of this pressure and the driving effect of *A*.

When the member *C* over-runs *A* then the rollers move to the left and disconnect the drive between *C* and *A*, in a similar manner to the action of the bicycle roller and wedge free-wheel.

The Rover type of free wheel shown in Fig. 144 operates upon this principle. In the diagram the various components are indicated, the gearbox drive being on the

left. In order to lock the free-wheel the toothed dog shown on the extreme left, which is a sliding fit on the splined gearbox mainshaft is slid along to the right so as to engage with the internal teeth of the outer free-wheel member. The control for this sliding dog is by means of a fork and Bowden cable arrangement with hand control wheel on the dashboard.

The principle of the Millam free-wheel is shown in Fig. 145. In this example the gear-box main shaft *B* has six radial

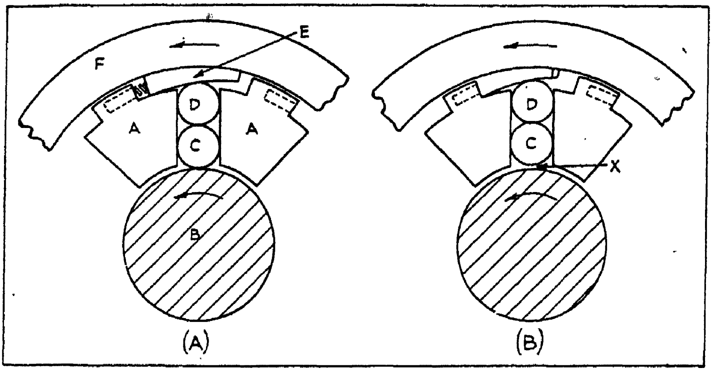


FIG. 145.—Millam Free-wheel Principle.

projections *A* integral with it. Between these members *A* are parallel slots in each of which a pair of rollers *C* and *D* are placed. The roller *D* normally bears on the wedge member *E*, the outer surface of which is curved to fit the inside of the driven annular member *F*; the latter conveys the drive to the propeller shaft when the engine is operating normally and propelling the car as in Diagram *A*. The member *A* has a projection at its outer end carrying a spring which pushes the wedge *E* to the right. The rollers thus become jammed between the shaft *B* and the ring member *F*.

When the car is over-running the drive, that is to say, when the car is descending an incline and the foot is taken off the accelerator the outer member *F* tends to turn faster than *B* and *A* so that the wedge moves to the left and the rollers, as a result of centrifugal action move outwards thus leaving a clearance at *X* (Diagram B) between roller *C* and the shaft *B*.

In this arrangement the stresses normally set up by the jamming effect of the rollers are minimised. A locking device is provided for reversing purposes.

Gear-Changing with Free-Wheel Devices.—As previously mentioned gear changing is simplified with the free-wheel. It is usually not necessary to employ the clutch once the car is in motion. Thus, the procedure for changing to a higher gear is as follows :—Accelerate the car and then take the foot off the accelerator pedal. Next, pause for a period of 1 to 4 seconds to allow the engine revolutions to decrease. Then move the gear lever over to the next higher gear position and accelerate. To change down the foot is lifted off the accelerator and at the same time the gear lever is moved directly to the next lower gear position. In this case there is no necessity to pause when making the gear change as when changing up.

If a very quick change down is required it is advisable to use the clutch in the usual manner.

An important point to remember for cars fitted with free-wheel devices is to ensure that the engine controls, *i.e.*, carburettor throttle or slow-running controls, are adjusted so that when the car is coasting the engine continues to 'idle' regularly ; unless this is done the engine may stop when the car is coasting.

An Automatic Clutch Control. An automatic device for operating the clutch so as to relieve the driver of a good deal of manual operation and skill which has been developed and fitted to some cars, is known as the Bendix clutch control.

The principle of this device (Fig. 146) is a very simple one. It utilises a vacuum-operated piston to depress the clutch-pedal when required. The vacuum cylinder and piston are connected by suitable piping to the inlet manifold of the engine, but between the cylinder and inlet manifold a control valve is placed. This valve is connected to the ordinary accelerator pedal, so that when the accelerator is depressed the valve is closed and the vacuum is shut-off. The vacuum-operated piston is thus cut out of action and the clutch is, therefore, released, i.e., engaged.

Whenever the accelerator pedal is released by the driver

the valve on the inlet pipe to the vacuum cylinder is opened and the piston operates, pulling the clutch out of its engagement.

Let us now see how this device affects the driving of the car. Firstly, *when the engine is started-up*, with the gear-lever in neutral, of course, the valve previously referred to is opened—for the accelerator pedal is free—and the piston, which is connected to the clutch by means of a cable attachment, therefore, depresses the clutch. The driver now *engages first* (*or lowest*) *speed* and accelerates the engine, thus cutting out the vacuum control and engaging the clutch.

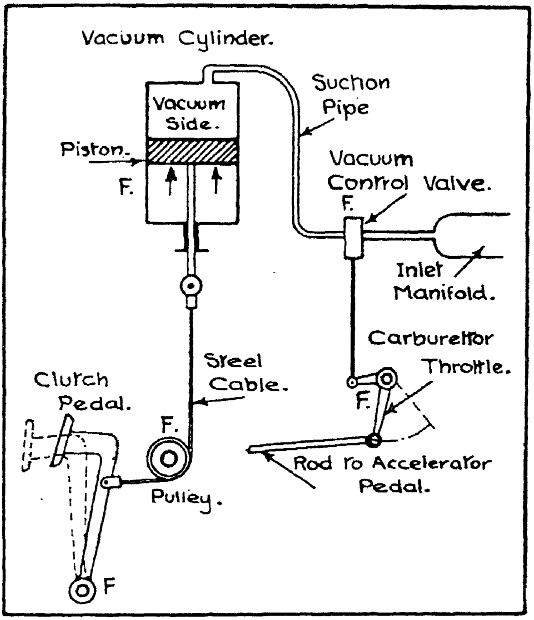


FIG. 146.—Principle of Bendix Automatic Clutch Control.

The same thing happens when each gear is engaged, until the top gear is reached, when the accelerator is kept depressed a certain distance, the clutch being, therefore, always engaged. Should the car be *running down a hill*, i.e., coasting, the accelerator is naturally released and the vacuum-controlled piston at once comes into operation, thus depressing the clutch and allowing the engine to idle.

Changing down is equally simple, but the gear-lever is left in neutral whilst the engine is accelerated to a higher speed to suit the lower gear-ratio. The accelerator is then released to allow the clutch to be automatically disengaged, and the gear-lever is moved into the next lower gear.

The important feature of the Bendix clutch control is that there is *no need for the driver to use his left foot on the clutch-pedal* at all, except when starting from rest. If the engine stops accidentally when coasting the vacuum is at once destroyed, the vacuum-operated piston, therefore, releases the clutch and the engine starts up again, for it is then 'driven' by the transmission.

In one form of the Bendix device the vacuum cylinder is on the engine side of the dashboard and its piston is connected by a steel cable passing over a pulley to the clutch-pedal. The vacuum control valve is merely linked up with the throttle lever on the side of the carburettor, so that when the throttle is shut the vacuum valve is open, but as soon as the throttle commences to open the latter closes.

Gillett Automatic Clutch Control. This control, adopted by the Borg and Beck Company, London, operates upon the hydraulic principle and holds the clutch out when the engine is idling for gear engagement, allows the clutch to be engaged in correct relation to the engine speed when the accelerator is depressed, gives positive engagement when the clutch is released and instant disengagement for gear changing purposes. Thus, immediately the engine is started at idling speed, the clutch is automatically released, the necessary gear can then be engaged, the accelerator depressed and the vehicle driven off with a smooth movement.

To change gear, the change speed lever is merely grasped and the action of so doing releases the clutch and enables the next gear to be selected. As the change speed lever is released, the clutch smoothly takes up the drive.

The essential features of the clutch control include *an oil pump* driven at engine speed; *a cylinder* formed as integral part of the unit fitted with *a piston* which is subjected to the oil pump pressure and when in operation throws the clutch out of engagement through a suitable external mechanism; *a centrifugal governor* driven off the oil pump shaft and *the*

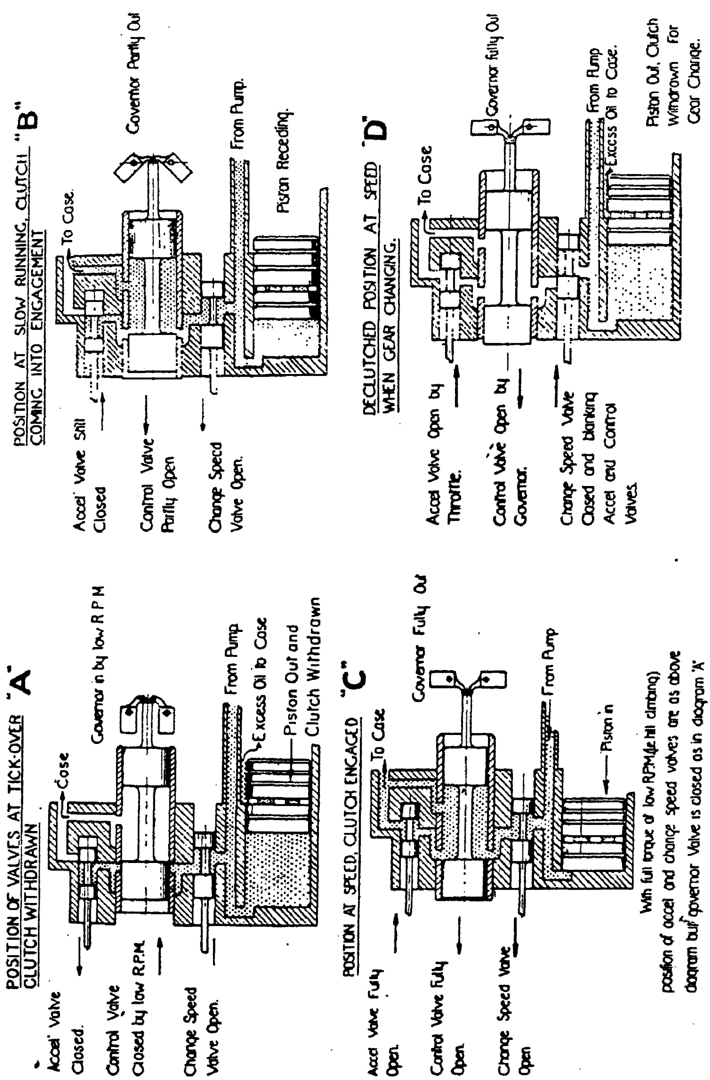


FIG. 147.—Illustrating Gillett Automatic Clutch Control Principle.

control valves. The latter are simple piston valves f directing the oil pressure to or relieving it from the cylind either automatically or at the will of the driver.

The various control valve positions are shown in Fig. 14 Diagrams *A*, *B*, *C* and *D*.

Referring to Diagram *A*, it will be seen that at idling speed of the engine, the oil pump has built up sufficient pressure behind the piston to throw out the clutch, and it should be noted that the centrifugal governor has not yet come into action owing to low engine speed. As the clutch is now released, the necessary gear can be engaged by the driver for starting the movement of the vehicle.

Diagram *B* illustrates a valve condition where the engine speed has been increased thus causing the governor to begin to open out. This permits the governor valve to partly open, so allowing the oil pressure behind the piston to escape, and consequently, the clutch to commence engagement.

It should be noted that in both Diagrams *A* and *B* the accelerator valve remains closed.

Diagram *C* illustrates the position of the valves when the engine is running at speed, with both the accelerator and governor valves fully open thus allowing the oil pressure from the pump to by-pass the piston so that the clutch remains engaged.

Diagram *D* shows the engine still at speed (accelerator and governor valves remaining as in Diagram *C*) but in order to throw the clutch out to change gear, the change speed valve is *closed*, so causing oil pressure to be built up behind the piston and the clutch to be released.

Compression springs are placed behind the governor valve which arrangement gives a cushioning effect upon the governor and can be adjusted to provide a rate of clutch engagement to suit individual engine characteristics.

The accelerator control valve is attached to the accelerator pedal by means of a flexible wire and arranged so that the valve is held closed when the accelerator pedal is in idling position. A coil spring gradually opens the valve when the accelerator pedal is depressed.

The change speed valve is held open by means of coil spring and is instantly closed at the driver's will by clasping the gear level knob. This incorporates a switch which is connected with an electrical solenoid used to close the change speed valve. Also when the change gear knob is released, the valve opens again thereby allowing the clutch to fully engage.

CHAPTER V

THE GEARBOX

The action and the manipulation of the gears in the gearbox frequently present difficulties to the beginner. It may perhaps assist the latter if it is stated that the petrol engine only gives its full horse-power at a fairly high speed, namely about 2,500 to 5,000 r.p.m. according to the type, and that it is not reversible in direction of rotation. Now the tractive force or effort required at the rear wheels to propel the car varies very considerably according to the conditions of use. Thus if the car is travelling at say 40 m.p.h. on a flat road, only a moderate effort, or turning-moment (*torque*) is required at the rear wheels, but at a high speed of rotation. When the car starts from rest a large torque is required at the rear-wheels, at a low wheel speed; this requires a fairly high engine output, and therefore high engine revolutions. Again when climbing a steep hill, a high torque is required at the rear-wheels, and maximum power and, therefore, engine revolutions. At the higher road speeds—60 to 80 m.p.h., say—a moderate torque but high engine revolutions are necessary. From these facts it will be seen that it is necessary to be able to vary the speed of rotation of the road wheels, whilst keeping the engine revolutions more or less constant.

There is another way of regarding this matter, namely, from the power-output and speed points of view.

The horse-power delivered to the road wheels depends upon the turning effort there and the turning speed.

Thus Horse Power $=$ Torque $\times$ R.P.M. $\times$ (a constant quantity). If the engine horse-power keeps constant, its speed also keeps constant. As, however, the resistance encountered by the car increases, a larger torque value is required; from the above relation it follows that a smaller r.p.m. of the wheels must result. If the resistance is small, then a smaller torque, and higher r.p.m. result. Since the engine speed is kept constant, it follows that some means

must be provided to vary the speed of the road wheels relatively to the engine. It is the purpose of the gearing in the gearbox to provide this variation, by introducing changes in the relative speeds, or ratios of the engine and propeller shaft, or back-axle. The ideal would be a gradually variable gear, but this is not yet possible, so that we have to be content with three, or four definite gear

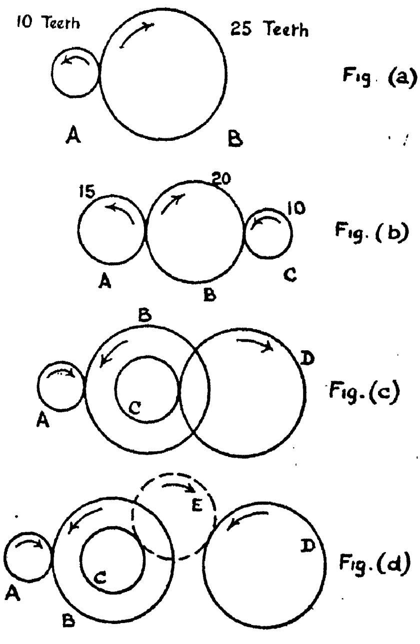


FIG. 148.—Principle of the Gears used in Gearboxes.

speeds at the most. To make up for this limitation, modern engines are sufficiently flexible to give their outputs over a range of engine speeds.

The Principle of Gearing.—When two gearwheels *A* and *B*, (Fig. 148 (*a*)) mesh together, their relative speeds of rotation depend upon their numbers of teeth. If *A* has 10 teeth and drives *B*, which has 25 teeth, *B* will rotate at $\frac{10}{25}$, or $\frac{2}{5}$ of the speed of *A*, but will have $\frac{25}{10}$ or $2\frac{1}{2}$ times the torque, or turning effort of *A* (neglecting the small loss of power through friction at the teeth). If *B* drives *A*, then

A will rotate at $\frac{25}{10}$, or $2\frac{1}{2}$ times the speed of B, but since the power depends upon the product of torque and r.p.m., the torque of A will be only $\frac{25}{10}$, or $\frac{2}{5}$ that of B. It is now easy to see how, for a given power input to one shaft, e.g., the engine shaft, one can vary either the torque or speed of another, e.g., the back axle, to suit.

It will be observed from Fig. 148 (*a*) that the wheel B rotates in the opposite direction to A. In order to drive another shaft, by gearing, in the same direction it is necessary to provide, either an intermediate wheel, namely B, in Fig. 148 (*b*), or two sets of gears, as shown in Fig. 148 (*c*). In this case A drives B, and a gear wheel C fixed rigidly to the same shaft as B, drives D. This latter arrangement is employed in automobile gear boxes, for the direct gears, using wheels of suitable ratios. For reversing purposes another wheel, such as (B) in Fig. 148 (*b*), is interposed between (C) and (D) in Fig. 148 (*c*), as shown in Fig. 148 (*d*);

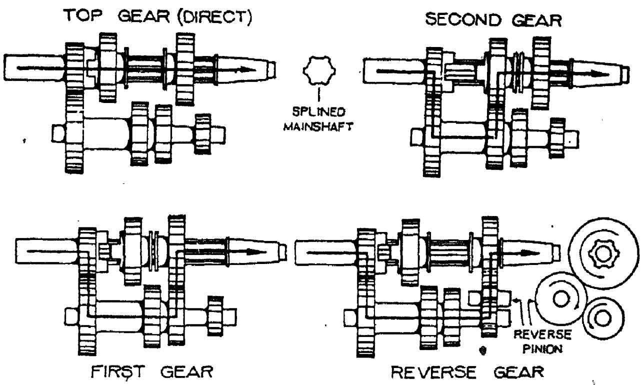


FIG. 149.—The Jowett Three-Speed Gear.

here the wheel E, is introduced between C and D. In actual gearbox practice the gears are not arranged in lengthy trains as shown in Fig. 149, but along two parallel shafts, some of the gears sliding for this purpose along the upper shaft.

Methods of Engaging Gears.—There are two alternative methods of engaging the gears in ordinary gear-boxes, namely (1) By means of dog-clutches and (2) By the use of sliding gear wheels.

In the former case one end of the sliding member has teeth which engage with corresponding ones on the fixed position rotating member as in the example of the top gear engagement illustrated in Fig. 149. It should be mentioned that, instead of simple dog clutches it is now the practice to use a pair of gear wheels, one cut with internal teeth and the other with corresponding external teeth; one of these members is arranged to slide along its splined shaft into engagement with the other fixed position one. By using a relatively large number of teeth on the gear wheels instead of the usual square or two tooth dogs a much easier gear engagement is obtained.

When sliding gears are employed, as in the example of the 3rd, 2nd and 1st gears shown in Fig. 149, it is usual to bevel the sides of the engaging gear-wheel teeth to facilitate their engagement.

A Typical Three-Speed Gear Arrangement.—The illustrations given in Fig. 149 show the application of the foregoing principles to an actual three-speed gearbox, viz., the Jowett one.

Referring to the top left-hand diagram, it will be seen that the dog-clutch on the left has been slid along the splined gear-shaft so as to engage with the corresponding dog member on the engine-driven shaft (on the left). The drive is then solid from the left to the cardan shaft on the right; this gives *top gear.*

In the top right-hand diagram the sliding dog on the upper splined shaft has been slid along to the right until its gear-wheel teeth have meshed with those of the second gear wheel (from the left) of the layshaft below. The drive is then through two pairs of wheels, as shown by the black arrow-line; this gives *second gear.*

In the lower left-hand diagram, the upper right-hand gear wheel has been slid along its splined shaft to the *left* until it meshes with the third wheel from the left on the layshaft; this gives *first gear.*

For the reverse gear the same upper right-hand gear wheel is slid to the *right* until it engages with the idle reverse pinion shown in the lower right-hand illustration. The drive is then through five wheels in all, as against four in the two preceding examples; the black arrow-line shows the line of the drive in reverse gear.

The Three-Speed Gearbox.—The three-speed gearbox is usually fitted to the majority of high-powered American cars and to certain mass-produced English ones—usually of the 8 to 10 h.p. class. It is less expensive to manufacture than the four-speed type and, with a good power to weight ratio car gives satisfactory results. Fig. 150 illustrates a typical three-speed gearbox, fitted with Timken roller bearings; the latter take both heavy loads and end thrusts.

The engine- (or more correctly, clutch) driven shaft *N*

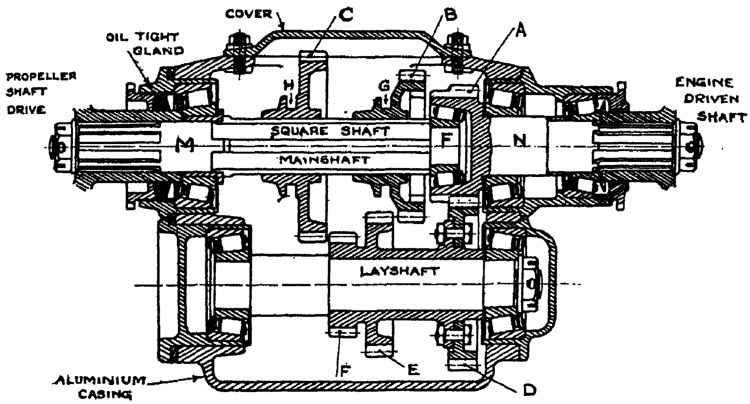


FIG. 150.—Construction and Working of the ordinary Three-Speed Gearbox.

is shown on the right; the propeller, or cardan-shaft drive *M*, on the left. The shaft *N* has a double gearwheel *A* formed on one end. The other gearwheels shown in Fig. 150 are *D*, *E*, *F*, *B* and *C*, of which the wheels *D*, *E* and *F* are formed solidly on or with the *Layshaft*, whilst *B* and *C* are capable of sliding along the square section shaft, or *Mainshaft M*. This shaft rotates at its right-hand side in a roller bearing *F*, formed in the hollow part of the gearwheel *A*, and so rotates quite independently of *N*. Gear changes are made by sliding either *B* or *C* into engagement with *A*, or *E*, and *F*, respectively.

Top, or Direct Gear (sometimes called 3rd Speed) is obtained by moving *G* and with it *B* to the right until its internal teeth mesh with the corresponding external teeth on the left of *A*. In this case the engine and propeller shafts are locked together and rotate as one shaft.

Second Gear, or Speed, is engaged by sliding G to the left until its teeth come opposite to, and mesh with those of E. The drive is then from A to D, and E to B. The speed of the shaft M is given by the relation (gear ratio):

$$\frac{\textit{Speed of } M}{\textit{Speed of } N} = \frac{\textit{No. teeth on } A \times \textit{No. teeth on } E}{\textit{No. teeth on } D \times \textit{No. teeth on } B}$$

Thus the ratio for $A = 8$, $B = 12$, $E = 8$, and $D = 14$ would be $\frac{8 \times 8}{12 \times 14} = \frac{1}{2{\cdot}625}$; this is termed a gear reduction of 1 to 2·625.

First Gear or Speed is given when C is slid along to the right to engage, or mesh with F. The drive is then from A to D, and F to C. It will be noted that the ratio of reduction $\frac{A}{D}\frac{F}{C}$ is much more. Thus if $F = 6$, and $C = 14$ we have for the ratio $\frac{8 \times 6}{14 \times 14} = \frac{1}{4{\cdot}08}$ The shaft M would therefore rotate at only about one-quarter of the speed of N, but it would give nearly four times the turning effect.

Reverse Gear is obtained by sliding another wheel having double width teeth (not shown, for reasons of simplicity) along, so as to mesh with both F and C; in this case F drives this *Reverse Pinion,* which in turn drives C, but in a reverse direction to the other three gear speeds.

Neutral, is the term applied when the shaft N, which is driven through the clutch from the engine, is not connected to M. When the car is left standing with its engine working, the gears are placed in neutral; this is the position actually shown in Fig. 150.

The gears B and C which give the three direct gear ratios are slid along the shaft M by means of semi-circular arms or *Strikers* which engage with collars G and H respectively. The rocking, or sliding motion of a single gear lever operated by the driver enables the necessary movements of the striking arms on G, H and the reverse pinion to be carried out.

The Four-Speed Gearbox.—In the case of many of the lower powered cars it has become necessary to employ four-speed gearboxes, in order to obtain better performances than with the three-speed type. Thus, there are many

occasions experienced in driving a car when it just fails to climb a hill on top gear, but would do so if a slightly lower gear ratio were available. With the three-speed gearbox the 2nd ratio gives too great a reduction under

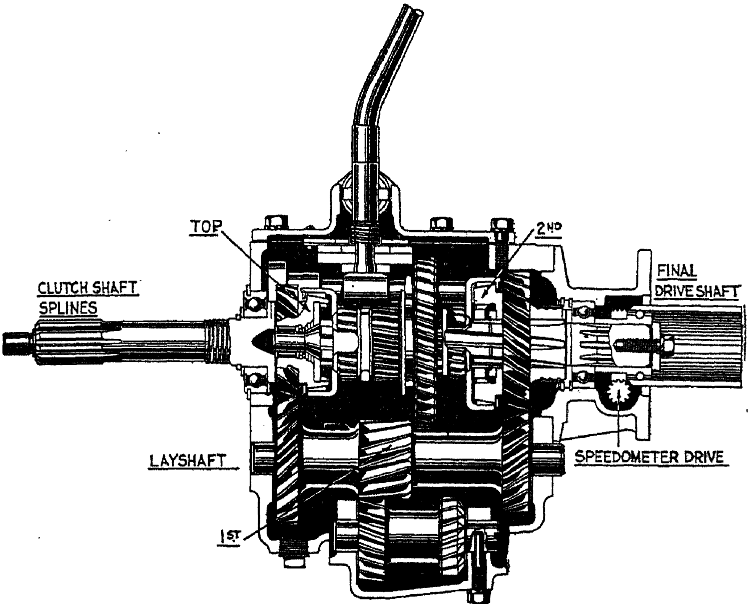


FIG. 151.—A Typical American Three-Speed Gear Box.

these conditions so that the car is forced to surmount the hill at a slower speed. The 3rd ratio of a four-speed gearbox—which is invariably higher than the 2nd of a three-speed one—enables the car to climb more rapidly. Usually the 2nd speed of the four-speed gearbox is arranged with a lower ratio than that of the three-speed one, so that except on inclines it is possible to start from rest in 2nd gear, using the 1st gear only for starting on inclines or with the fully laden car.

Fig. 152 illustrates a commercial pattern four-speed gearbox, in which the gears are slid along the splined primary shaft into engagement with their mating members.

The engine drive is taken through the clutch unit to the smaller gear *A* which is in constant mesh with the larger layshaft gear *B* ; the layshaft therefore revolves at a lower speed than the engine crankshaft in the same ratio as the

number of teeth on *A* bears to the number on *B*.

To engage 4th (top) gear the wheel *D* is moved to the left so that its internally cut teeth mesh with corresponding

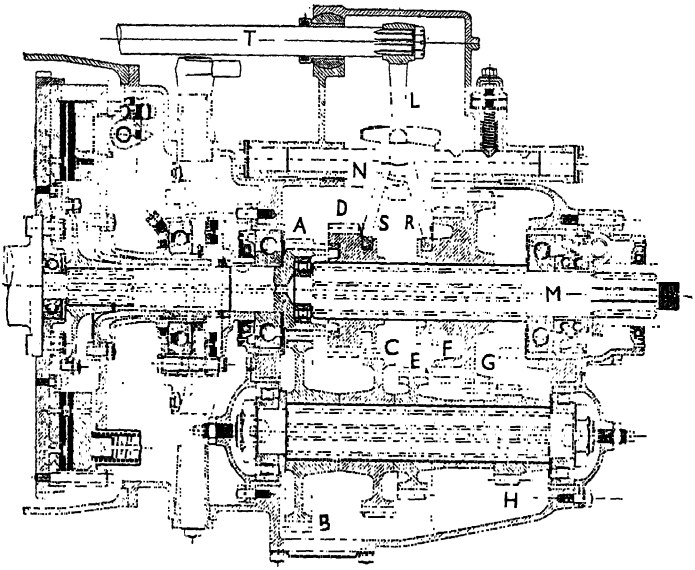


FIG. 152.—Four-Speed Gear Box.

external teeth formed on the right of *A*. The drive is then taken direct from the engine to the main shaft *M* and thence to the propeller shaft and back-axle.

The 3rd gear is obtained by sliding *D* to the right in order to mesh with the gear wheel *C*; the drive is then from *A* to *B* and from *C* to *D*.

The gear ratio is thus $\frac{\mathrm{A}}{\mathrm{B}} \times \frac{\mathrm{C}}{\mathrm{D}}$ where the letters represent the corresponding numbers of teeth on the gears concerned.

The 2nd gear is obtained by sliding the gear *F* along the splined shaft *M*, to the left until it meshes with the gear *E*. The gear ratio is then given by $\frac{A}{B} \times \frac{E}{F}$.

Finally, 1st (bottom) gear is engaged by sliding gear *G* to the right into mesh with the small gear *H*. In this case the gear ratio is given by $\frac{A}{B} \times \frac{H}{G}$

The reverse gear is obtained in a similar manner to that of the three-speed gearbox, using the reverse idler wheel.

In regard to the operation of the sliding gears on the splined shaft *M*, which rotates independently of the clutch shaft, the gear lever moves the shaft *T* in a backward or forward direction and a ball-end on the lever *L* engages with one or other of the selector or striking arms *S* or *R* according to which gears are to be engaged. The shaft *T* is rocked from one side to the other to engage either *S* or *R*. When *S* is engaged the 4th and 3rd gears can be obtained; when *R* is used then the 2nd and 1st gears are available.

The striking or selector rod *N*, it will be observed, slides in bearings at its two ends and is retained in the positions corresponding to the complete meshing of the respective gears by means of a spring-loaded plunger having a tapered lower end which slips into one or other of the notches in the rod *N*. Pressure upon the gear lever, when gear changing is sufficient to force the rod to slide out of engagement with the plunger. The main shaft *M* runs in deep race ball bearings and has a double ball thrust washer to take end loads; the layshaft runs in roller bearings. End adjustment of the layshaft is possible by means of the screws shown.

Gear Ratios.—Apart from the speed ratios given by the gears in the gearbox, and which give for the top speed, direct coupling with the engine, or clutch shaft, there is another fixed reduction gear, usually between 4 to 1 and 5 to 1 in the final drive in the back axle, so that when the gear lever is in top gear, the back axle rotates at a $\frac{1}{4}$ or $\frac{1}{5}$ engine speed; it is usual in stating gear ratios to include the back axle reduction, so that top gear would be stated as being 4 to 1 or 5 to 1, as the case may be. In the case of three-speed gear boxes for 12 h.p. cars, the following are typical ratios employed: 1st, 15 to 1; 2nd, $7\frac{3}{4}$ to 1; 3rd, (top), $4\frac{3}{4}$ to 1; Reverse, 19 to 1; evidently the back axle reduction is $4\frac{3}{4}$ to 1 in this case. In 'bottom' or 1st gear the road wheels rotate at $\frac{1}{15}$ engine speed, but give about 15 times the turning effort.

In the case of a typical four-speed gearbox, the ratios employed are: 1st, 18 to 1; 2nd, 11 to 1; 3rd, 7 to 1, and 1st, $4\frac{1}{2}$ to 1; Reverse, 20 to 1.

Gear Changing.—The art of changing gears correctly with plain gear boxes lies in the proper adjustment of engine speed, and road wheel speed, with a judicious use of the clutch.

To change 'up,' that is from bottom gear to top, as when starting from rest, with gear lever in neutral, the clutch is disengaged by depressing the clutch pedal as far as it will go; this is termed *declutching;* the gear lever is placed in bottom (4th or 3rd) gear position, and the clutch is engaged gently by releasing gradually the clutch pedal. As soon as the clutch begins to engage, gradually accelerate the engine by depressing the foot accelerator—not too much, however—when the car will glide forward, gathering speed. In the usual case, when a road speed of from 3 to 4 m.p.h. is attained, declutch, move the gear-lever into the next higher gear (3rd or 2nd) and *afterwards* let in the clutch

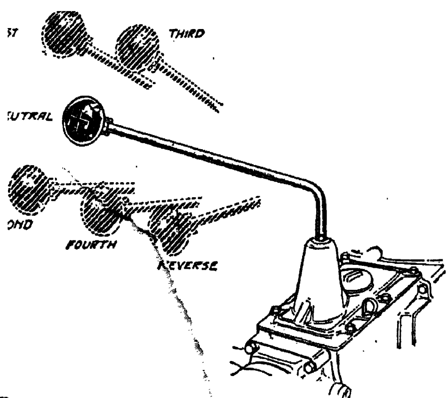


FIG. 153.—The Standard Four-Speed Gear Lever Position.

gradually. Speed up the car to say, 6 to 10 m.p.h. repeat the operation for the next gear. It advantageous to pause for a moment, in the travel, when moving the gear lever into the next position, before pushing the lever over.

To change down. This is more difficult as a rule, and is one of the operations regarded with disfavour motorists. To change down from top to nex as when slowing down on a hill or in traffic, known as *Double Declutching* is the

silent change. In this method, the clutch pedal is depressed, gear lever placed in neutral, (half-way between top and next gear) ; releasing the accelerator whilst this is done the clutch is next engaged, and the engine accelerated by depressing the accelerator ; the clutch is then disengaged, and the gear lever moved into next lower gear. Finally the clutch is engaged smoothly. With a little practice, changing down is quite easy ; it is best to memorise the consecutive operations thus : Declutch, Gear Lever Neutral,

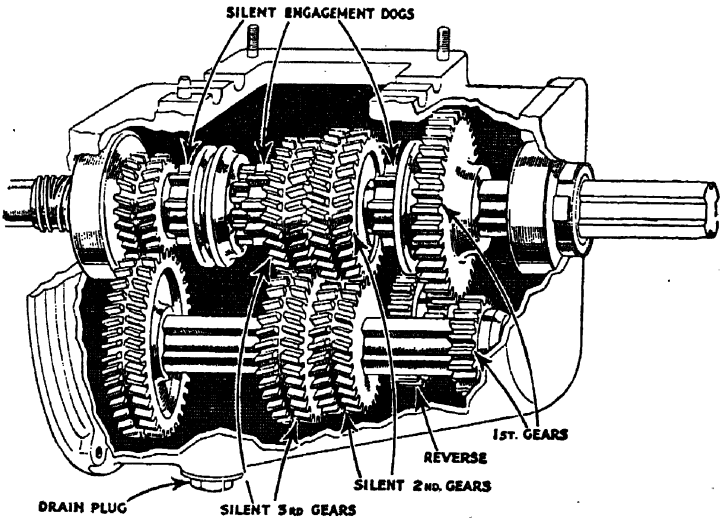


FIG. 154.—Four-Speed Gear Box with Pairs of Helical Gear Wheels for Quiet Operation and to Balance End Thrusts (Rover).

Engage Clutch, Accelerate, Declutch, Gear Lever 2nd (say), Engage Clutch. The object of the intermediate acceleration is to speed up the layshaft, so that its gear wheels are moving at the same speed as the mainshaft when the gear-wheel on the one is slid along into mesh with the corresponding wheel on the other ; it is clear that the two wheels will only engage when they are both rotating at the same speed. The whole principle of gear-changing consists in speeding up one or other of the shafts, until they both rotate at the same speed.

If a mistake is made, in gear changing, declutch immediately, and apply the brakes ; start the car from rest again. Never try to force the gears into mesh, or the

edges of the teeth will wear down quickly and serious damage may result.

Gearbox Position.—The modern tendency for some time has been to bolt the gearbox to the engine casing directly; this gives what is known as *Central Unit Construction*, as shown in Figs. 115 and 151. It is particularly suited to mass-production, dispenses with intermediate flexible drives, and gives a neat, clean appearance. On the other hand it is usually difficult to get at the clutch for adjustment, and dismantling of the gearbox is difficult.

In some cases the gearbox is placed some distance behind the engine, and an intermediate shaft provided with flexible couplings (or universal joints) transmits the drive from the engine through the clutch to the gearbox. These universal joints are a necessity since it is not possible to fix the engine with its crankshaft axis dead in line with the gearbox shaft; further any flexure of the chassis frame due to road shocks, or loads, may cause a relative movement of the two shafts.

Sometimes the gearbox is held at four places on two cross-members of the frame; sometimes by the three-point suspension system with rubber block mountings similar to those now used for the engine. Previously, in certain cases the gearbox was built integrally with the front end of the propeller shaft, and moved up and down with it as the rear springs deflected; in this case there was a large ball joint formed on the front of the gearbox. In other cases the gearbox was arranged at the rear end of the propeller shaft casing, i.e., just in front of, but integral with, the back axle; in this case there was an increase in the unsprung weight.

Holding the Gears in Mesh.—In the case of norma designs of three- and four-speed gear boxes after the gear lever has been moved so as to engage any particular gear it is necessary to maintain the gears in mesh, i.e., to prevent them from jumping out of engagement. The usual method of ensuring this is to provide notches on the end of the sliding selector gear bar, such that when each of the gears is engaged a spring loaded plunger with suitably shaped end drops into the notch (Fig. 152). The sides of the plunger end and notch are inclined to permit of disengagement

when required; alternatively a spring-loaded ball is employed instead of a plunger. Each gear lever 'engaged' position has its corresponding selector rod retaining notch and plunger or ball.

Fig. 155 shows the Vauxhall selector mechanism in two positions. The upper view indicates the gears engaged;

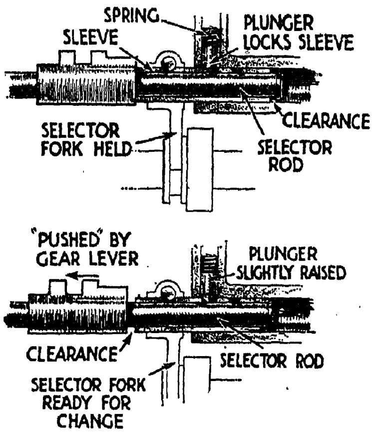


FIG. 155.—Gear Locking Mechanism.

the fork by which they are operated is then prevented from moving because the sleeve in which it is carried is held by a plunger fitting into a hole. The lower view shows what happens when the gear lever is moved towards the 'neutral' position. The selector rod is then caused first to slide within its sleeve, thus taking up the clearance between a collar at the end of the rod and the flat surface at the end of the sleeve. This small movement pushes the parallel portion of the plunger clear of the hole so that the cone-ended plunger can then jump clear of the hole in the sleeve.

Gearbox Care and Attention.—There is seldom anything to go wrong, provided the gearbox is kept well lubricated. The layshaft should be half immersed in the lubricant recommended by the manufacturers. Too much lubricant, it has been shown, causes an absorption, or loss of power. Once every 8,000 to 10,000 miles or so the gearbox should be drained, by removing the plug in its base; the interior

should be cleaned out with paraffin, well drained, the plug replaced and fresh lubricant put in.

Trends in Gear Box Design. The earlier 'crash' change type gear box required a certain amount of skill in order to make a quiet gear change. With the wide-spread use of motor cars much attention was given to the use of alternative devices and improvements to simplify gear changing. Modern gear change systems include the use of epicyclic gears, synchro-mesh devices, free-wheel units, fluid couplings, electric- and vacuum-operated gear change mechanisms, overdrives, automatic variable transmissions requiring no gear levers and semi-automatic systems. The general tendency in the latest models is to dispense with the ordinary gear change lever and to employ a miniature gear lever and quadrant or 'gate' on the steering wheel column for selecting the required gear. The actual gear changing operation is performed by automatic means, such as the movement of a piston actuated by vacuum from the inlet manifold; electro-magnetic clutches for engaging epicyclic gears are also employed.

Typical examples of some of the more recent transmission systems are described in this Chapter.

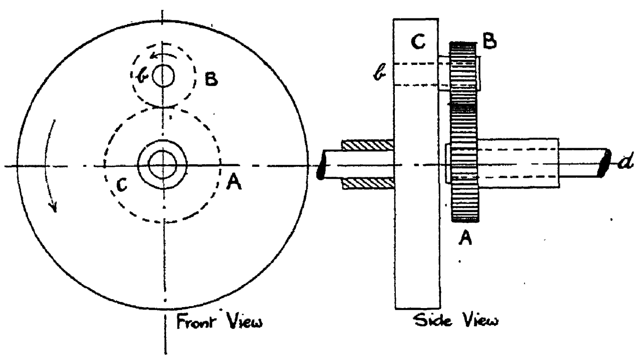


FIG 156.—Principle of the Sun and Planet (or Epicyclic) Gear.

The Epicyclic Gear.—The Lanchester and the Ford cars were hitherto fitted with a speed gear known as the *Epicyclic* or Sun and Planet. Several modern car trans-

missions now employ the epicyclic principle. The special feature of this gear is that there are no sliding dogs, or gears to engage, the operation of gear-change being performed by merely tightening brakebands on the gear drums; this renders gear changing very simple.

The principle of the epicyclic gear is illustrated in Fig. 156, in which *C* represents a flywheel carrying the pin for the small gear-wheel, or pinion *B*'s bearing *b*, the gear-wheel *B* meshes with the gear *A*, which can rotate about its shaft *d*.

First imagine the gear wheel *A* is fixed rigidly, and the flywheel *C* to rotate. Then the pinion *B*, which is free to turn on the pin *b* will rotate, in virtue of its teeth meshing with those of *A*. If the numbers of teeth on *A* and *B* are 32 and 8 respectively it can be shown that for every revolution of *C*, *B* will make $\frac{32}{8} + 1 = 5$ revolutions relatively to *A*; the extra revolution is due to the pin *b* having been carried around once.

Next suppose that *A* is free to rotate on its shaft *d*, but that *B* is locked solidly to *C*. Then one revolution of *C* will cause *A* to turn in the same direction one revolution; in fact the gear *A* will be locked solidly with *C*; this arrangement could be employed for a high gear, or direct drive in a car.

A little consideration will show that if by some external means *B* can be rotated quickly, whilst *C* rotates slowly in the directions shown by the arrows, the gear wheel *A* can be made to rotate in the opposite direction; this arrangement will give a reverse gear. By varying the speed of *B* we can vary both the speed and also the direction of *A*.

Having obtained an elementary idea of the properties of the sun and planet system, we will pass on to an actual example of a two speed and reverse gear system, viz., shown diagrammatically in Fig. 157.

In this case there are three sets of pinions *B*, *D*, and *E*, cut out of the solid, which can rotate on the pin *b*. These mesh, respectively, with three sets of independent pinions or gear wheels *A*, *F* and *G*, arranged to rotate one within the shaft of the other as shown. The numbers of teeth on *B*, *D* and *E* are 27, 33 and 24 respectively; those on *A*, *F* and *G*, 27, 21 and 30 respectively. The wheel-shafts *F* and *G* are provided with drums *H* and *J* upon which

band brakes can act, so as to stop their motions when required. There is a brake-drum *K* on the extension of the shaft of gear-wheel *A*. It will also be observed that the flywheel spigot, or extension shaft forms the central bearing shaft of *A*, and that a series of clutch plates are keyed to it. It will be apparent that when the clutch plates on the flywheel shaft are allowed to engage with those fixed to the drum *K*, of the shaft of *A*, that both *A*, *C* and *d* rotate solidly. This gives the *high gear*.

The *low gear* is obtained by holding the clutch plates out of engagement, and applying a brake to lock the drum *J*; these operations are both obtained by pressing the low

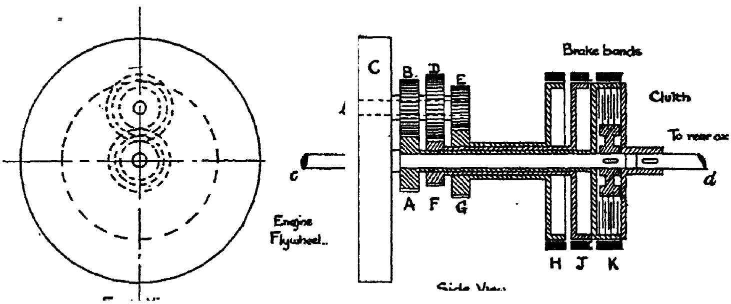


FIG. 157.—Principle of the Two-Speed Epicyclic Gear.

gear pedal. When the drum *J* is locked, the wheel *F* is also locked, and as the flywheel *C* carries *B* and *D* around with it, *D* is caused to rotate more slowly (actually $\frac{12}{33}$ of a revolution), and *B* which is also rotated at this speed, gearing into the larger wheel *A* causes it to rotate at the same speed, since *A* and *B* have the same number of teeth. Thus the shaft of *A*, which is connected to the back-axle, through the drum *K* will rotate at $\frac{12}{33}$ $(\frac{1}{2\frac{3}{4}})$ of the flywheel's speed; this gives a gear reduction of $2\frac{3}{4}$ to 1, to which, of course must be multiplied the gear ratio of the back-axle, to give the total reduction at the rear wheels.

The reverse gear is obtained by applying, by means of the reverse pedal the brake *H*, which then locks *G*, the drum *J* running idle; the clutch is also disengaged as before. As *C* rotates, the pinion *E* is caused to rotate in the same direction but at a somewhat higher speed, owing to the number of teeth on *E* being 24, whilst those on *G*

are 30. The result is, that A is turned in the reverse direction at a speed of $\frac{30-24}{24} = \frac{1}{4}$ that of the flywheel, thus giving a reverse ratio of 4 to 1.

For starting the engine, it is arranged that the *hand-brake lever* in its mid-travel position operates the clutch lever, and holds the clutch out, thus disconnecting the rear-wheel drive from the engine. It should be repeated that when either brake is applied to H or J, the clutch is automatically disengaged, and that only one drum can be locked at a time, the other being idle.

The De Normanville Transmission System.—An interesting modern example of the epicyclic gear principle is that employed upon the Humber cars; it is known as the De Normanville system. This comprises a four-speed

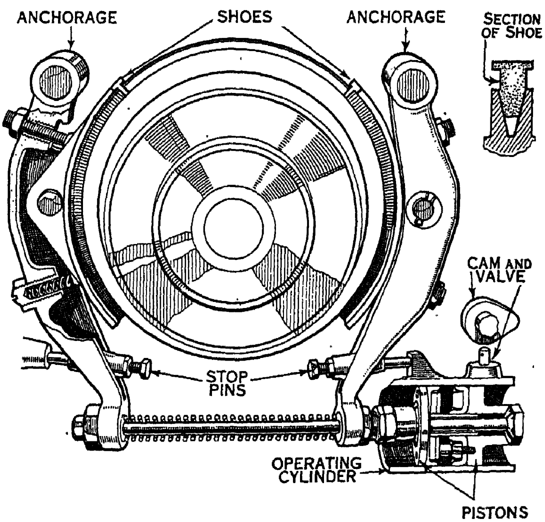


FIG. 158.—The Brake Shoes and Oil-Operated Pistons.

and reverse gear with an additional device for free-wheeling or coasting. An ordinary type of plate clutch is fitted between the engine and the gear box. The usual gear lever is dispensed with and in its place a small gear lever with quadrant is mounted on the upper end of the steering column, conveniently for the driver's operation.

Gear changing is carried out by first depressing the

ordinary clutch pedal, in order to disconnect the transmission from the engine. The small hand gear lever is then moved to the position (marked on the quadrant) corresponding to the required gear ratio. Upon releasing the clutch gradually, the selected gear is engaged smoothly.

The principal feature of this system is the method of locking the epicyclic gear brake drums, there being three separate drums and epicyclic gear trains ; top gear consists of a direct cone clutch connection between the driven main plate clutch member and the driven gear shaft which is connected to the propeller shaft. Each brake drum (Fig. 158) has a pair of brake shoes which are suitably hinged and brought into engagement with the drums by means of a pair of pistons in a cylinder to which oil pressure is applied. In Fig. 158 the two pistons are shown in the

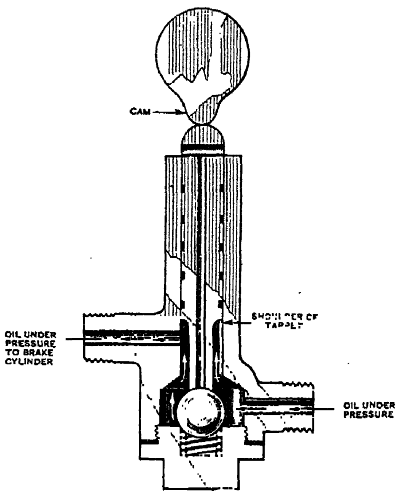


FIG. 159.—Operating Cam and Valve.

braking positions, having been forced outwards by oil pressure applied in the cylinder space between them.

In order to apply any brake so as to engage a selected gear, the operation of the gear lever is arranged to rotate a camshaft (Fig. 159), which causes a plunger to lift a spring-loaded ball valve off its seating thus placing the brake cylinder of the selected gear into communication with a source of hydraulic pressure. The latter consists

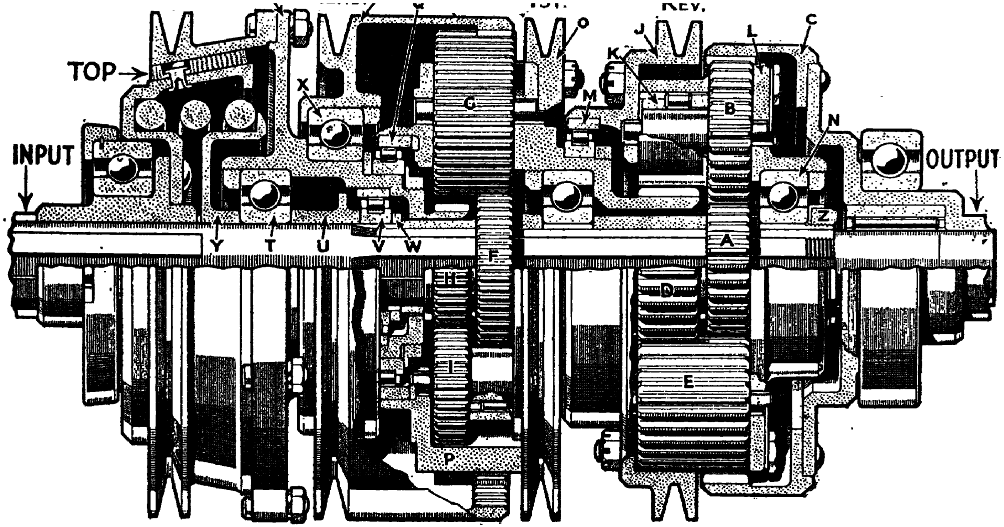


FIG. 160.—The De Normanville Four-Speed Gearbox.

KEY

A.	Reverse Sun Wheel.	J.	Reverse Drum.	S.	2nd Speed Annulus.
B.	Reverse Planet Wheel.	K.	Planet Wheel Bearing Distance Collar.	T.	3rd Speed Brake Drum Cover Bearing.
C.	Tail Shaft Annulus.	L.	Reverse Drum Cover or Planet Carrier.	U.	Distance Piece.
D.	2nd Speed Drum Casing and 1st Speed Sun Wheel.	M.	1st and Reverse Drum Bearing.	V.	3rd Speed Drum Cover Roller Bearing.
E.	1st Speed Planet Wheel.	N.	Reverse Drum Cover Bearing.	W.	3rd Speed Drum Cover Roller Bearing Distance Piece Washer.
F.	2nd Speed Sun Wheel.	O.	1st Speed Drum.	X.	2nd Speed Drum Bearing.
G.	2nd Speed Planet Wheel.	P.	2nd Speed Drum Cover or Planet Carrier.	Y.	Top Speed Spring Support.
H.	3rd Speed Sun Wheel.	Q.	2nd Speed Drum Cover Bearing.	Z.	Mainshaft Lock Nut.
I.	3rd Speed Planet Gear.	R.	3rd Speed Brake Drum Cover and Sun Wheel.		

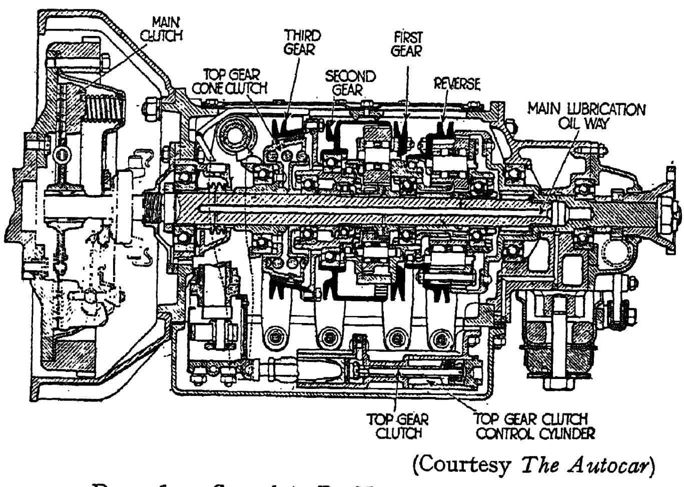
of a reservoir filled with oil to which pressure is applied by means of a plunger type pump located in the gearbox and driven by worm gearing from the gearbox main shaft. The reservoir has a pressure release device so that when a pre-determined pressure is attained any surplus pressure causes the release valve to uncover a port, allowing the oil to escape into the gearbox.

Each of the 1st, 2nd and 3rd gear brakes has its own separate hydraulic operation means for the brake shoes.

The coasting position of the small gear lever corresponds to a 'neutral' position of the gears, above the top or 4th gear position. When in neutral all the gears are idle, the brake shoes being clear of their drums and top gear disengaged.

Tests of this method of transmission, using a Heenan Highfield electric dynamometer have shown that it has an efficiency of 99 per cent on 3rd gear and 98 on second.

Constant Mesh Gearboxes.—In order to avoid having to slide, and to mesh the gear wheels one into the other, some designs of gearbox arrange for the gears to rotate



(Courtesy *The Autocar*)

FIG. 161.—Complete De Normanville Gear Box.

freely upon their own shafts, or bearings, and to always be in mesh with the other gears on the other shaft. In place of the sliding gearwheels, separate sliding collars, with

teeth—sometimes termed '*dogs*'—are moved into engagement, so that whichever gear is engaged, rotates with the squared or castellated shaft driving the 'dog.' One particular type of gearbox used to employ silent chains to drive the opposite pair of gears.

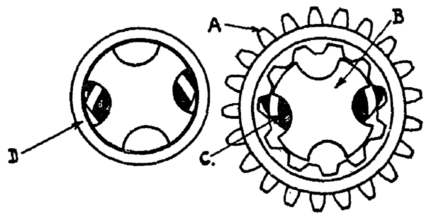


FIG. 162.—Chandler Simple Gear Change Method.

In the Chandler gearbox, the gears are always in mesh, and one of the pairs that are required to drive is locked solidly to the driving shaft by means of special keys. In Fig. 162 the keys *C*, are shown held in the free position by means of a metal bearing ring *D*. These rings are pressed on the shaft and revolve with it, forming the bearing on which the gears run. To engage a gear the keys are slid endwise, so as to clear the bearing ring; the keys then rock on their longer axes, and engage with notches cut on the inside circumference of the gearwheel *A*, as shown in Fig. 162 (right hand diagram).

Silent-Second and Third Speed Gears.—Most modern types of gearboxes are fitted with intermediate gears which run practically as silently as the top gears; the latter, it will be remembered are 'direct' gears which operate as though the engine-shaft were connected solidly to the back axle.

In the case of a three-speed gearbox, having what is known as a 'silent-second' speed, the gears giving the latter ratio are usually of the constant-mesh type, i.e. always in mesh as before stated. These gears normally run freely on their shafts, and are brought into operation by means of sliding dogs on one of the shafts in question.

It is a well-known fact that it is easier to engage a dog clutch than to slide one gear into mesh with another one which is invariably running at a different speed—as in the case of the ordinary gearbox.

In the 'silent-second' gearbox the dog clutch method is used to engage the middle gear instead of sliding the gear along a splined main-gear shaft, as in ordinary practice. If these constant-mesh gears are of the well-known helical or double-helical (or herringbone) type, they will always run silently together.

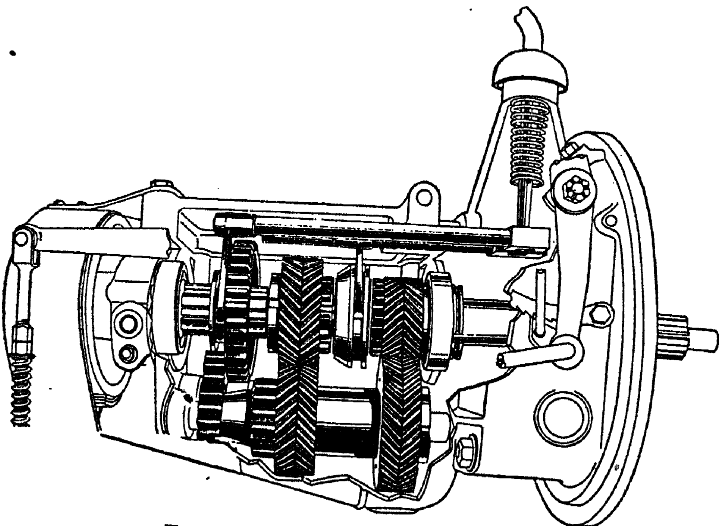


FIG. 163.—The Reo Silent Gearbox.

In four-speed gearboxes the third speed—which is next in ratio to the top gear—is frequently of the constant-mesh type; it is then known as the 'silent-third' speed.

Some makers also apply the constant-mesh principle, not only to the next highest ratio to top gear, but also to the other gears.

In the case of certain four-speed gearboxes not only the third, but the second and first speed also are of the 'silent' type. Fig. 154 on page 173 shows one type of Rover four-speed gearbox in which the 1st, 2nd and 3rd gears are of the double-helical constant mesh pattern; the gears are engaged by the sliding dogs shown.

The Warren Synchronizer.* This device is fitted to the rear end of the gear box as shown in Fig. 164 and it is

* Pulsometer Engineering Co., Ltd., Reading.

operated directly from the clutch pedal, so that when the clutch is disengaged the gear box as a whole is disconnected from the engine in front and from the propeller shaft behind. It, therefore, quickly comes almost to rest and gear changing is simply a matter of moving the lever into whichever gear is required. It is not necessary to exert any pressure on the gear lever, as there are no cones or friction devices to operate, as with synchromesh.

Having engaged the desired gear it is only necessary to accelerate, when the synchronizer will be automatically locked.

The device does not entail free-wheeling, the drive being automatically locked after every gear-change, thus enabling full use to be made of the engine as a brake. If, however, *the driver wishes to free-wheel* on any particular hill, it is only necessary to depress and release the clutch pedal when the car will free-wheel until the accelerator is again depressed, when the synchroniser will again lock up automatically.

Referring to Fig. 164 the synchronizer consists of a free-

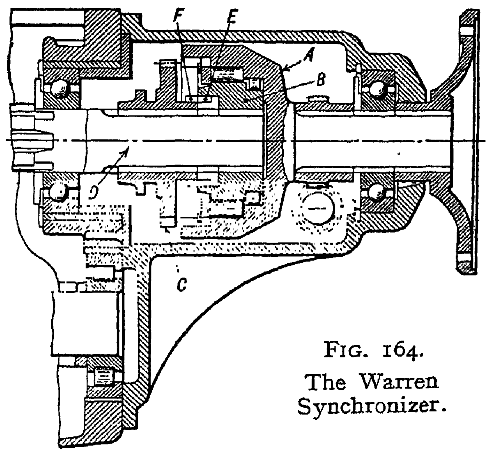


FIG. 164.
The Warren Synchronizer.

wheel member *A* and *B*, and a locking dog *C*, which slides on the end of the gearbox shaft *D*, and is coupled up by a rod to the clutch pedal so that when the pedal is depressed it is withdrawn from engagement with the member *A*, thus disconnecting the gearbox from the propeller shaft and allowing an easy gear-change to be made. The member

B is free to move slightly on the shaft *D*, so that its teeth *E* balk the teeth *F* on the locking dog and prevent the dog engaging with the member *A* until the gearbox is accelerated up to a car speed and the teeth *E* and *F* are tripped, thus allowing the locking dog *C* to engage with the member *A*. When reverse gear is engaged the locking dog is automatically moved into engagement with the reverse dogs on the synchronizer.

The Synchromesh Gear.—Introduced a few years ago into this country on certain cars made by Messrs. General Motors, Detroit, U.S.A., this method of easy gear changing is now practically universal in all gear boxes.

It consists, in principle, of a mechanical device for synchronizing the speeds of the two gear wheels or dog-clutch members which have to be engaged with one another,

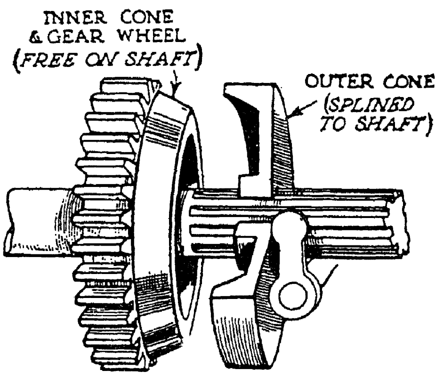


FIG. 165.—Showing how the Inner and Outer Cones are arranged to Lock the Gears to the Splined Shaft.

before the actual meshing operation takes place. When both the members are running at the same speeds gear changing can be accomplished with both ease and silence. The manner in which the synchromesh device works is illustrated, in an elementary manner, in Fig. 165. This shows the two clutch members, the right-hand one of which can slide on the splines, whilst the left hand one is free to rotate on the plain end of the splined shaft.

The first movement of the gear lever brings the outer cone (driven indirectly by the engine) into contact with the

inner cone, so that the two revolve at the same speed. Further movement of the gear lever moves the gear wheel, shown in Fig. 166 into mesh with another internally cut gear inside the driven shaft gear wheel, thus engaging

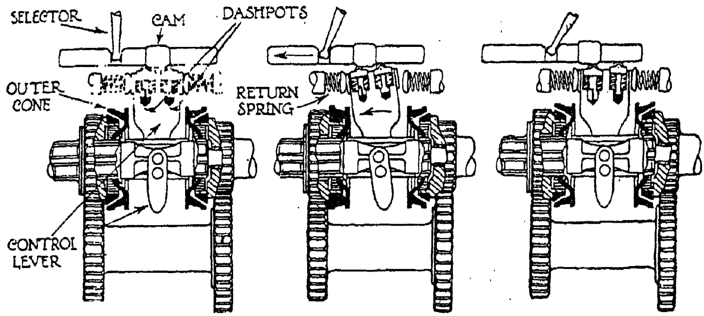


FIG. 166.—Three Stages in the Changing of a Gear, with Synchromesh Cones. (*The Autocar*).

the gears smoothly. When these two gears are engaged a spring device disengages the outer cone and returns it to its initial position.

In order to make the action as smooth as possible oil dash pots are usually provided on the gear lever selector mechanism near the cone operating device.

The common arrangement for synchromesh gears is for the change from 2nd to 3rd gear, and from 3rd to 4th (top) gear in the case of four speed gearboxes. For this purpose two sets of synchromesh cones are fitted, as shown in Fig. 166.

It will be observed that the synchromesh principle is employed for constant mesh gears, so that by making the latter of helical or herringbone design very quiet running is ensured.

The original synchromesh gear systems have been improved upon in order to overcome one or two faults that were found with these, notably a tendency to noisy operation when the gear lever was moved too quickly; it was thus possible to 'beat the cones' by using undue pressure on the lever.

Later Improvements.—The manner in which the earlier synchromesh gears have been improved is well illustrated in the instance of the more recent Vauxhall gearboxes;

with these it is impossible to produce noisy gear change by undue pressure on the gear lever. Figs. 167, 168 and 169 show the essential parts of the gear change in the simplest form.

Referring to Fig. 167 the initial movement of the gear lever results in light contact between the cones, the outer

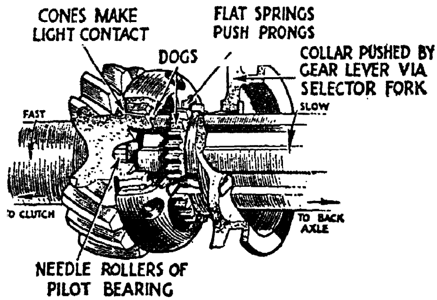


FIG. 167.

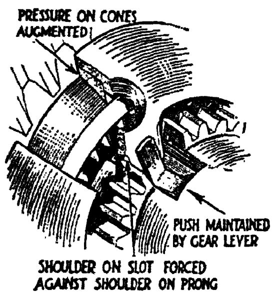


FIG. 168.

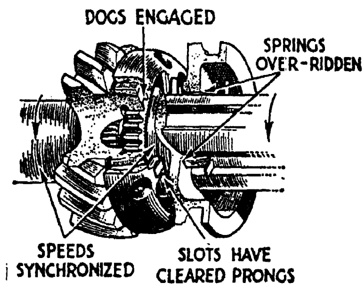


FIG. 169.

one of which is pushed towards the inner cone by three flat springs. Contact pressure between the cones is augmented (Fig. 168) by the forces exerted between the shoulders formed on the prongs and slots. This pressure ensures the rapid synchronizing of speeds and also prevents further movement of the gear lever until synchronization is complete.

As soon as the cones, dogs and shafts are all turning at the same rate (Fig. 169) the pressure between the shoulders is relieved so that they no longer prevent the dogs from moving into engagement. This movement

ermitted by slots in the sliding member which clear prongs of the outer cone.

A popular method used for the synchromesh mechanism of cars, including the Morris, Ford and Austin employs a series of spring-loaded balls in the inner cone member. The latter is made in two parts, namely, an inner annular member containing the internal cone and an outer concentric component that can slide along the inner member upon overcoming the restraint of the spring-loaded ball. The outer member contains the dog that is to be meshed with the corresponding part on the gear to be engaged.

Fig. 170 shows the Austin synchromesh unit that operates upon the principle mentioned. The inner member *C*

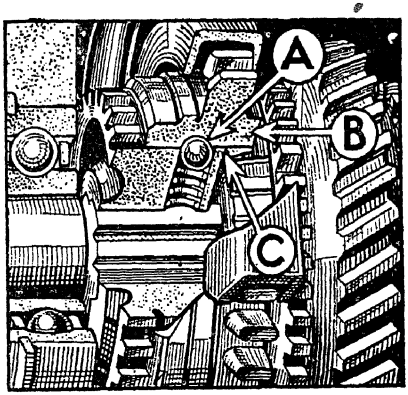


FIG. 170.—The Use of Spring-loaded Balls in Synchromesh Gears.

carries the internal cone (shown on the right) whilst the outer member *B* contains the dogs or teeth to be meshed with the gear on the right. One of the spring-loaded balls is shown at *A*. When the gear lever is first moved both members *B* and *C* slide towards the right so that the synchronising cones engage and bring the gears to the same speed. Further pressure on the gear lever forces the outer member *B* to the right—overcoming the resistance of the ball *A*—so as to engage the gear.

Types of Synchromesh Gearboxes.—When first introduced it was usual to provide synchromesh gear change between

the top and next highest gear ratio. Subsequently, owing to the successful appeal of this easy-change mechanism, many cars extended its use to the next gear

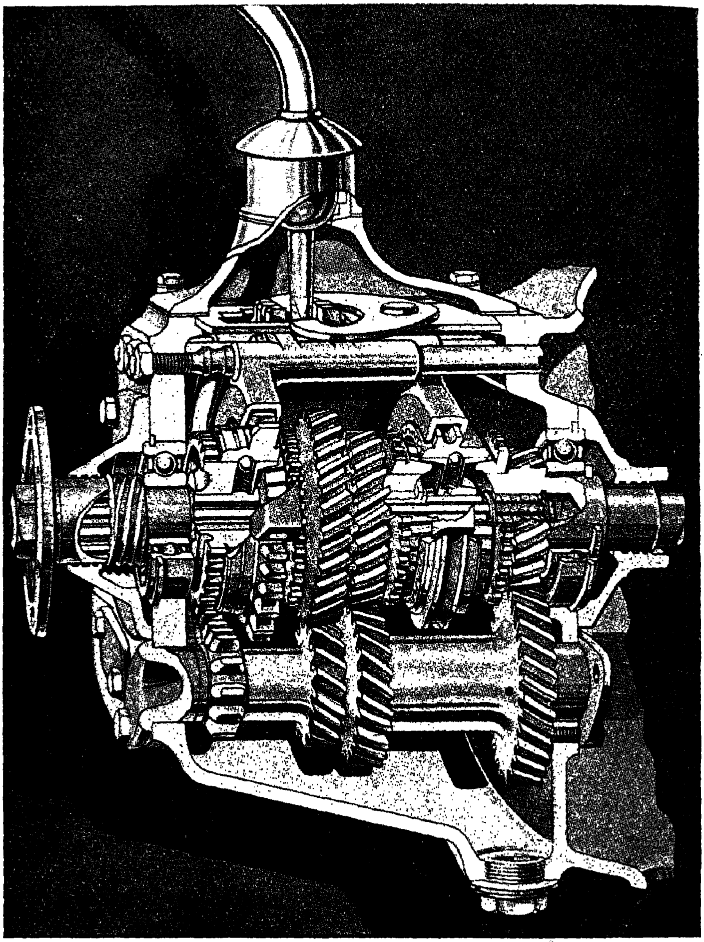


FIG. 171.—The Austin 10 H.P. Four-Speed Synchromesh Gearbox.

ratio, so that with four-speed gearboxes there was a synchromesh unit between 4th and 3rd and 3rd and 2nd, leaving only the bottom gear as a 'crash' change. The

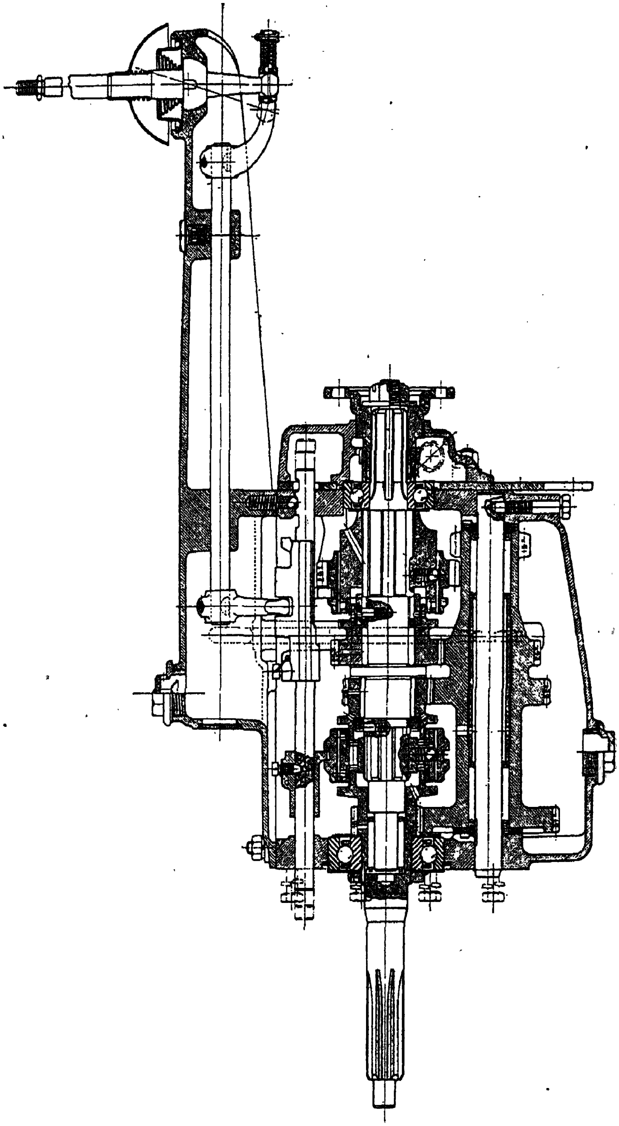


FIG. 172.—The Riley 12 H.P. Car Four-Speed Gearbox with Remote Control Gear Lever.

Austin 10 H.P. car synchromesh four-speed gearbox shown in Fig. 171 provides for the easy-change synchromesh principle to the 2nd, 3rd and 4th (top) gear ratios. The

second speed synchromesh unit is embodied in the sliding first speed gear. The third and fourth synchromesh mechanism is incorporated in the gear coupling.

In certain of the more expensive cars having four speed gearboxes it is now the practice to provide synchromesh devices for all four gears. A typical example of this is the Alvis all-synchromesh with double-helical silent gears of the constant mesh pattern.

An example of a remote hand-control gearbox is given in Fig. 172 which illustrates the Riley 12 h.p. pattern. This is a four-speed model having syncromesh on 4th, 3rd and 2nd gears. The gearbox is placed well forward on the chassis, so that it is necessary to extend the gear control as illustrated in order to bring the gear lever into a convenient position for the driver.

The main shaft is mounted on large deep-row ball bearings and the lay shaft is anchored to the gearbox casing; the secondary gear unit, having four integral gear wheels rotates upon this fixed shaft. The position of the selector rod and its spring loaded ball stop device are clearly shown in the illustration. The gearbox has a filler cap above and another plug at the base for oil emptying purposes.

Twin-Top Speed Gears.—Although the silent-third speed gear is often used with a little higher gear ratio than in normal practice, this leaves a marked step between the third and second gears, so that a big drop in speed occurs on changing down to second. A novel method used on certain modern cars is to fit what is known as a 'twin-top' gearbox. In this case the third gear ratio is brought much nearer the top gear, so that it is only about 25 per cent. to 30 per cent. below top ratio. It is used mostly on high-powered cars, which employ their top gears practically all the time. The twin-top gear enables a car to employ the higher top gear ratio for the open road, and the lower top gear ratio for town and traffic work.

The twin-top gear differs materially from the 'silent' gears previously mentioned, in its design and operation.

Fig. 173 illustrates the principle of the Graham Paige twin-top gear. In this case the *engine*-driven or clutch-shaft is shown on the left. The drive is taken from this through a toothed gearwheel *A* which, for third gear

position, is slid along to the right so as to engage with corresponding internally-toothed wheel *B*; in effect, this is equivalent to a simple dog clutch, but with a greater number of dogs or teeth. The gear *B* has external teeth which engage with internal teeth *C* on a drum arranged eccentrically with the axis of the main gearshaft. At the

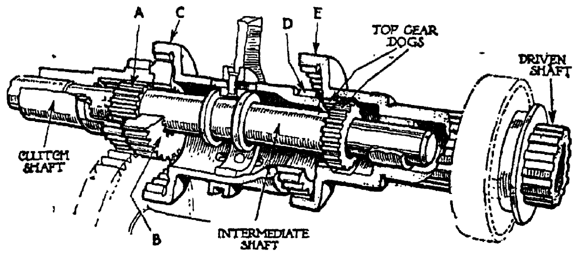


FIG. 173.—The Graham Paige Twin-Top Gear Arrangement.

other end of this drum is another gearwheel *D* that meshes internally with the teeth of the wheel *E*.

The third gear drive is, therefore, through the two gearwheels *B* and *C*, and thence through the two gears *D* and *E* to the driven shaft. When it is desired to engage top gear the gear *A* is slid to the left, and at the same time the shaft carrying *A* also slides the top gear dogs (shown on the right of this shaft) into mesh with internal teeth on the gear ring *E*. It should be noted that *E* is integral with the final drive, or driven shaft, shown on the right.

The Overdrive Gear System.—In effect this arrangement is similar to the 'twin-top' gear one, but instead of having a driver's control for engaging the higher speed gear this operation is performed automatically—usually by centrifugal action. The overdrive, in most cases, is engaged when the car speed reaches or exceeds about 40 m.p.h. and it usually cuts out again at a rather lower speed, namely, at about 35 m.p.h.

In many American cars the controlled overdrive system is employed. This gives a combination of ordinary or direct drive below about 25 to 30 m.p.h., the overdrive being engaged automatically on releasing the accelerator above this speed. In order to return again to the direct drive the accelerator pedal is depressed to its full extent

at any speed above about 25 m.p.h.; the free wheel inoperative during this procedure.

that a choice
erdrive system.

The overall overdrive ratios in the case of high powered ars lies between 3·1 and 3·5, the rear axle ratios being 4·3 to 4·8.

The advantages of being able to maintain high road speeds with the overdrive gear engaged are the same as those for the 'twin top' gear car, namely (1) Lower engine revolutions for a given road speed and therefore less wear and tear on the engine and gearbox, and (2) Lower fuel and oil consumption per mile.

In most overdrive systems there is a small gearbox arranged at the back of the main one, c an epicyclic type of gear change to give the r gear ratio reduction from 'top' to 'overdrive top'

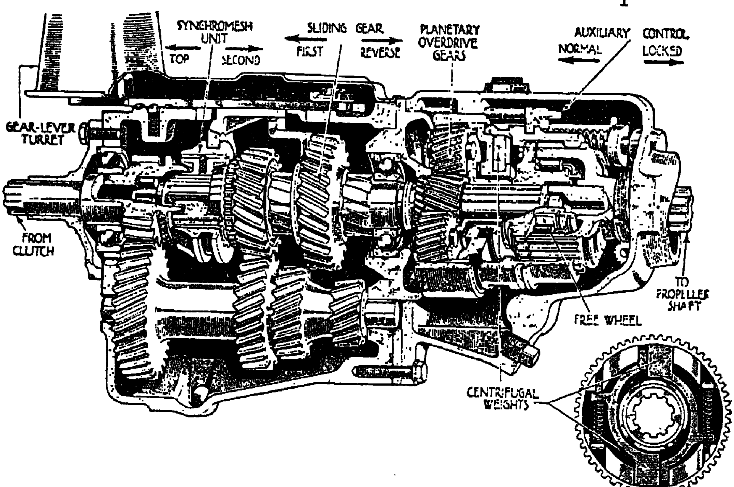


FIG. 174.—The Riley Overdrive Gearbox with Free-wheel Unit.

Fig. 174* illustrates the Riley Nine gearbox which has three normal forward speeds and embodies a free-wheel unit. The overdrive engagement is effected by the centrifugal action of two weights dependent

*Reproduced by permission of *The Motor*.

for their operation upon the road speed of the car; when the latter exceeds about 43 m.p.h. the weights move outwards and lock one of the epicyclic gear elements thus bringing the gear into operation.

In regard to the gearbox all of the gears are of the helical pattern; synchromesh engagement is provided on the top and second gear. Motion is imparted through the constant mesh gears of the primary shaft to the gearbox layshaft. Through this medium the drive is obtained for the indirect gears, that is to say, first and second gear. Second gear is also in constant mesh but except when in use, the gear on the mainshaft freewheels. With the selection of this gear the dog clutch of the synchronizing unit, which is splined to the mainshaft, engages the gear and the drive is then imparted through the layshaft to the mainshaft. First gear is a sliding gear which is brought into operation by shifting the gear on the mainshaft into engagement with the layshaft gear. Top gear is engaged in a similar manner to second gear except that the synchronizing unit is moved in the opposite direction.

The overdrive consists of a set of planetary gears which rotate round a sun wheel. When driving in normal gear the drive passes through the centre or sun wheel of the overdrive, the planetary gears engaging with the outer gear ring. With the engagement of the overdrive, centrifugal weights move outwards to engage with slots in the outer gear ring, with the result that the planetary gears no longer idle round the sun wheel but are driven round with it. The effect of this is to alter the high gear to ratio to 3·97 as against 5·5 on traffic top.

A roller type of freewheel is incorporated at the rear of the gearbox. As the name implies the freewheel 'rolls into' and 'out of' operation. The parts comprise an annular ring with cams ground on its periphery and encircled by a steel cage. Rollers are fitted into the grooves of the cage and the splined driving unit.

The Warner overdrive unit which has been used extensively on American cars is illustrated in Figs 175 and 176.

The epicyclic action of this auxiliary gear is exactly similar to the Wilson, the sun pinion being the stationary member, and the pinion carrier the driver which causes the annulus to revolve at a higher speed than the mainshaft of the gearbox.

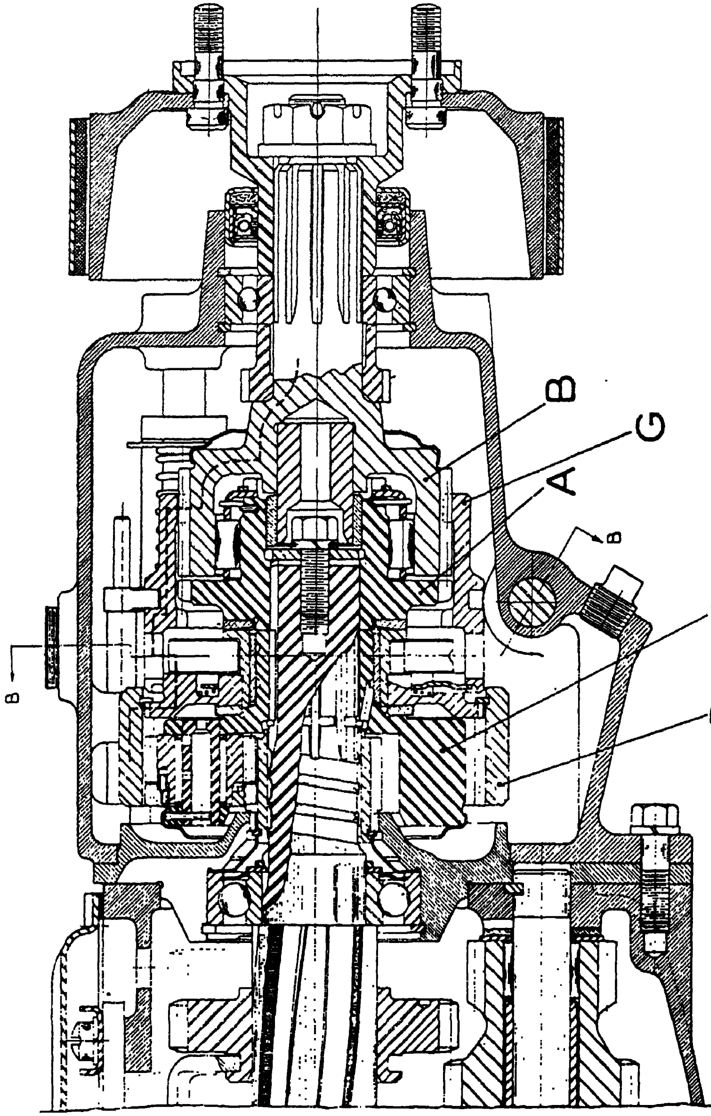


FIG. 175.—The Warner Overdrive System. (Side Section.)

On direct drive the freewheel hub A is integral with the mainshaft and power is transmitted through the rollers direct to the output shaft B; as the pinion carrier C is also integral with the same shaft and the sun pinion is

permanently held stationary, epicyclic action also takes place, but the annulus D is free, revolving at overdrive speed. When a certain speed is reached the sliding dogs E, carried in a housing fixed to the annulus, tend to move outwards under the influence of centrifugal force against the light resistance of springs F. On release of the accelerator the outer sleeve G, driven by the output shaft,

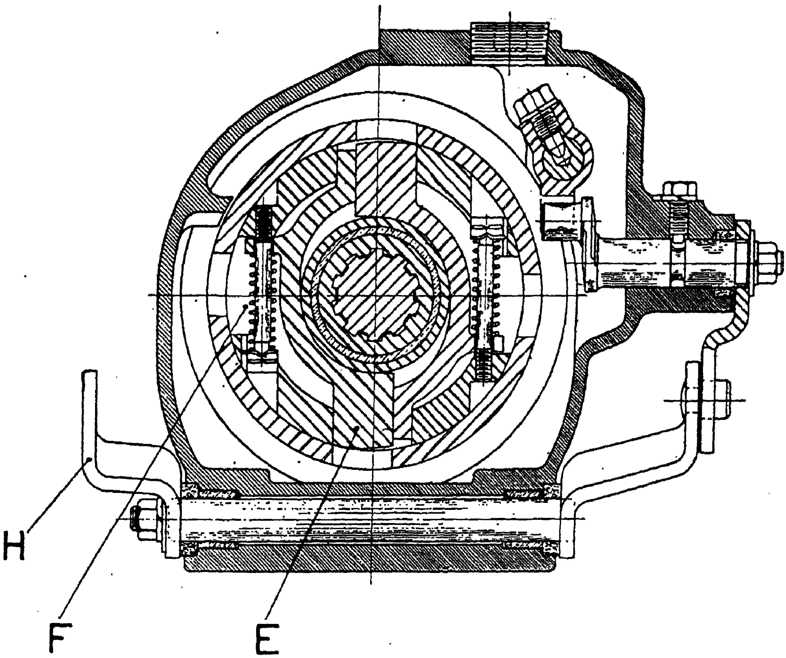


Fig. 176.—The Warner Overdrive. (Cross Section.)

creeps on the dog carrier until the dogs come into line with the slots with which they engage, thus connecting the higher speed (or overdrive) of the annulus to the output shaft as shown by the dotted line.

The freewheel does not operate on overdrive. The overdrive and freewheel can be cut out of operation by actuation of lever H (Fig. 176), which slides sleeve G, the teeth of which engage with A, locking it to the output shaft.

The epicyclic gear train is in operation continuously and the annulus D is therefore revolving at overdrive speed under all conditions. This device is efficient, silent

running, and useful on cars of suitable power/weight ratios, particularly in countries having long, straight, and comparatively flat roads.

Non-Automatic Overdrive System. There is one drawback to the automatic overdrive method, namely, that it does not permit the same acceleration and deceleration rates, for high speed road driving purposes of the ordinary gearbox. In order to overcome this disadvantage it is necessary to provide an independent control for the driver's use, so that the overdrive can be brought into or out of engagement at will.

A representative design of overdrive of this type is the de Normanville which embodies a hydraulically-operated epicyclic gear unit.

The Chrysler Semi-Automatic Underdrive Transmission. This four-speed semi-automatic transmission is used on recent Chrysler and de Soto cars in conjunction with a fluid flywheel unit and a conventional clutch. It provides four forward speeds and two reverse ones. The gear lever has three positions, namely, 'High,' 'Low' and 'Reverse.' Normal driving is done with the gear lever in the former position, so that starting, cruising and stopping conditions are thus possible owing to fluid drive and semi-automatic type of transmission provided.

With the gear lever in the high position the transmission can be moved back and forth from third gear (1·55 : 1) to fourth gear (1 : 1), in the same manner as the shift is made between third speed and overdrive in cars so equipped. That is, the car is started in third gear and stays there until the driver releases the pressure on the accelerator pedal when the car speed is over 15 to 17 m.p.h. At this time the transmission is moved to fourth gear by the action of a vacuum diaphragm unit connected to the engine manifold. When a burst of speed or extra power is needed, pushing the accelerator past the full-throttle position energises a solenoid which acts to shift the transmission back to third gear by admitting atmospheric pressure behind the diaphragm of the vacuum unit (provided that the car speed is below 53 m.p.h.). When the car speed drops below 13 to 15 m.p.h. with the transmission in fourth gear, the

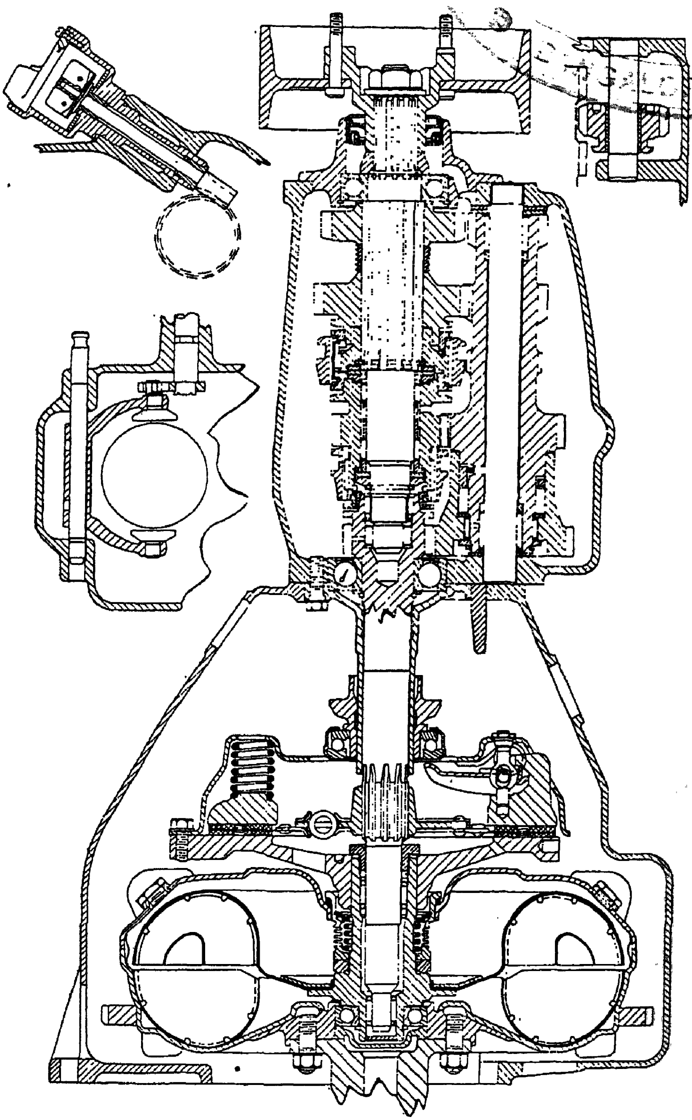


FIG. 177.—The Chrysler Semi-Automatic Underdrive Transmission.

position of the centrifugal governor causes the solenoid to be energised and the transmission to be shifted automatically back to third gear. A lower rear axle gear ratio than is used with conventional transmissions—either

3·54 or 3·73 : 1—is used in conjunction with the transmission.

Use of the low position of the gear lever is recommended only for steep hills, etc. When operating in this range, shifts between first and second are made in exactly the same way as are those between third and fourth with the gear lever in the high position. The change to second gear can be made when the car speed has reached 8 m.p.h. It can be stepped down to first again, when the speed is below 27 m.p.h., by depressing the accelerator past the full-throttle position. The ratio in first gear is 3·07 : 1; in second gear, 1·98 : 1.

The single-plate clutch is used in the usual way only for reverse and when the gear lever is shifted. The transmission is provided with a free-wheeling unit which is operative only in first and third gears.

Two reverse ratios are available—3·25 : 1 and 2·10 : 1. The second ratio can be obtained when the car is above the governor cut-in speed by momentarily releasing the foot from the accelerator pedal. The transmission free wheels when coasting in the 3·25 : 1 ratio reverse speed, but not in the second reverse speed.

The driving unit and housing of the fluid flywheel is bolted to the engine crankshaft as shown in the illustration, and the driven unit is connected to the drive pinion of the 'underdrive' transmission through the conventional clutch shown. The drive gear on the countershaft is fitted with a free-wheeling unit. The transmission itself comprises essentially three sets of constant-mesh helical gears, and a reverse set of spur gears. A synchronising clutch at the left engages when a power shift is made from first to second, or from third to fourth speeds, and disengages to return them. The manual synchroniser (at the right) is operated by the gear lever to cause the power to flow directly out to the tail shaft in the high position, or through the low-speed gears in the low position.

The function of the control system is to cause engagement and disengagement of the synchronising clutch. The vacuum unit assembly is mounted on the right side of the transmission and contains the 'kick-down' solenoid and the ignition interrupter switch. The governor assembly (in the upper right view in the illustration), containing the governor switch, is mounted on the right

side of the transmission case. The kick-down switch is prevented from closing when the engine is above a certain speed by the action of the carburettor venturi vacuum on a small plunger in the switch.

An emergency lock-out cable is provided for locking up the synchronising clutch and placing the transmission in the non-free-wheeling second or fourth speeds for starting the engine by towing the car.

Double Reduction Gears.—Another method sometimes adopted for extending the range of gear ratios available is to fit, in addition to the ordinary gearbox of three or four gear ratios, an entirely separate gear-unit of the two-speed type with its separate gear control. Thus, for each ratio of the ordinary gearbox there are two alternative ratios in the supplementary gearbox, so that there are now *six* speeds available instead of three in the case of the three-speed gearbox, and *eight* for the four-speed gearbox.

There was a supplementary two-speed gearbox fitment marketed a few years ago for the commercial Ford car; in this case a two-speed gearbox was fitted on the propeller-shaft and back axle unit.

The Lagonda, Mercedes and Voisin cars have used these supplementary gearboxes.

Certain commercial vehicles, also, have a supplementary two-speed gear device in the back axle casing.

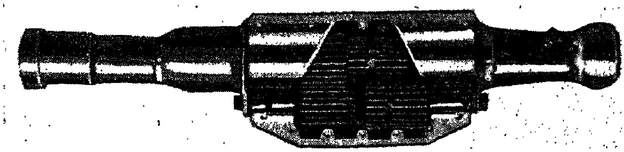


FIG. 178.—Suction-operated Voisin Double Reduction Gear.

In the Voisin arrangement, the auxiliary gearbox is fitted in the propeller shaft casing and the gears are operated by engine suction controlled by a small lever conveniently situated on the dash. The gears, in the case of the 16 h.p. Voisin car, give the following ratios: sixth,

or top gear, 4·7 to 1; fifth, or *low top speed*, 6·7 to 1; fourth, or *high second*, 7·8 to 1; third, or *low second*, 11·4 to 1; second, or *high first*, 14 to 1; and first, or *bottom low gear*, 21·3 to 1.

The ratio of the two gears in the auxiliary gearbox is approximately 1·4 to 1, so that there is 40 per cent. difference in speed and torque on each of these gears.

The pinions in the auxiliary gearbox are made of exceptional width and are chamfered to give easy engagement.

In connection with the operation of the auxiliary gearbox the gear selector lever for this is coupled to a piston which is capable of a semi-rotational as well as a reciprocating motion. Thus, the piston twists one way as it travels forwards, and the other way when moving backwards. A natural movement is thus given to the selector. Into the cylinder containing this piston are led two pipes connected through a valve on the dashboard to the induction manifold. It will thus readily be seen that by turning the small lever on the dashboard controlling the valves, one way or the other, suction can be exercised on either side of the piston. Moreover, the smooth cushioned effect of the engine suction prevents the severe clashing of the teeth of the pinions when gear is being changed.

While this arrangement allows a change to be made from the lower to the higher set of speeds regardless of the rate at which the car is travelling, the change down between the two series of ratios can only be effected at moderate speeds, such as below 20 m.p.h. on top gear, or corresponding engine speeds on the lower gears.

Pre-Selective Gearboxes.—An important development in gearboxes, which is now fitted as standard to Daimler and Armstrong-Siddeley cars, is that embodying the Wilson pre-selective gear principle.

In this case the ordinary gear-lever is dispensed with and its place is taken by a small lever mounted on the steering-wheel in a similar manner to the usual spark advance lever. This 'gear control' lever can move in an arc to marked positions corresponding to 'reverse,' 'neutral,' 'first,' 'second,' 'third' and 'fourth' speeds.

To change from one forward gear to another above or below it, the control-lever is moved by hand to the required gear position marked on the steering-wheel sector. No

effort is required for this operation and nothing happens in regard to the gear changing until one depresses the gear-change pedal and then releases it, when the selected gear is automatically engaged. One can, therefore, pre-select the next gear at any convenient time before actually changing to this gear.

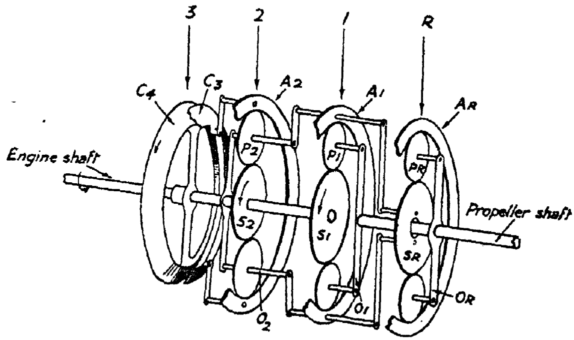


FIG. 179.—Principle of Wilson Three-Speed Gearbox.

The action of this gearbox is somewhat complicated, but the following is a brief outline of its operation:

The gearbox in question is a compound epicyclic gearbox giving (in addition to a 'solid' drive on fourth or top speed) three forward speeds and one reverse speed. All gear-wheels are permanently in mesh. Each indirect gear is obtained by applying a band brake of special form to one of four drums, each of which is integral with one element of a simple epicyclic gear.

The arrangement of the three-speed gearbox is shown in Fig. 179. There, the elements have been spread out and the number of planet pinions has been reduced from three to two for the sake of clarity.

The principal elements of the epicyclic gear are also shown in Fig. 180.

Neglecting, for the time being, the reverse train R, the essential driven member is the cage O_1, carrying the planets of the first gear train. The sun wheel of this train S_1 is integral with the engine shaft. If the annulus A_1 is brought to rest by a brake (the other brakes being free) the gear ratio is that of the train $S_1 — A_1$. This gives *first speed*.

Imagine next that the brake on A_1 is released and that on A_2 applied. The sun wheel of this train S_2 is also integral

with the engine shaft. The planet cage O_2 then rotates at a fraction of the speed of the engine in the same direction. It carries with it the annulus A_1 to which it is permanently

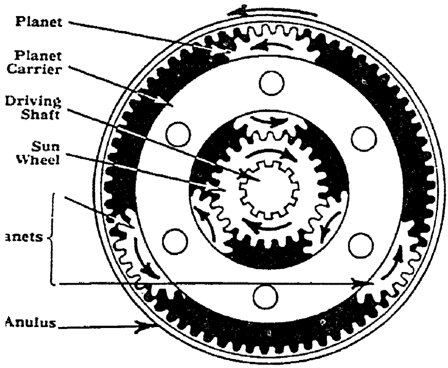


FIG. 180.—Principle of the Epicyclic Gear used on Pre-Selective Gearbox.

connected. The result is to increase the speed of the planet cage O_1, thus giving a higher gear, viz., *second speed.*

For high speed (direct) all the brake bands are released and the clutch $C3$ (integral with $A2$) is engaged with $C4$ (integral with the engine shaft). Thus $S2$ and $S1$ are running at engine speed.

The inter-connections between second and first speed

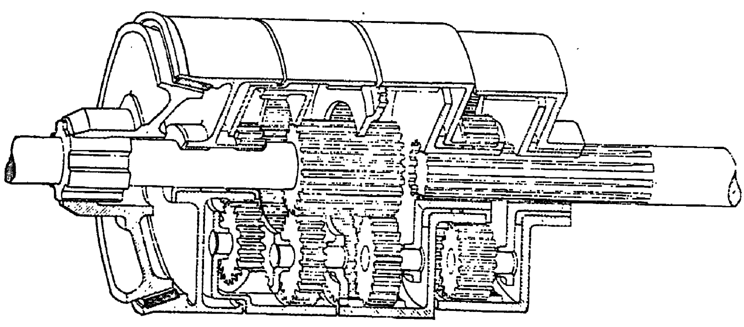


FIG. 181.—A Pictorial View of the Wilson Pre-Selective Gearbox.

trains (annulus of one to the planet carrier of the other) is such that no rotation is possible except as a solid block at engine speed. Hence A_1, the first speed annulus, runs at

engine speed. Since both annulus and the sun of the first speed train are now at engine speed, the planet cage and propeller-shaft are compelled to follow. All gears are locked.

Reverse is best considered by imagining the propeller-shaft to rotate and considering the consequences. It carries both O_1 and O_R. A_R if fixed by a brake, A_1 is free, but coupled to S_R. Hence S_R, carrying with it A_1, advances in the same direction as the propeller-shaft at a faster speed. Thus the annulus A_1 overtakes the planet cage O_1, resulting in S_1 (the engine-shaft) rotating in the reverse direction.

From the above explanation, together with an inspection of Fig. 179, it will be gathered that whenever the engine is running all the gears are in relative motion (excepting when the third speed clutch is engaged, when the gear is 'solid'). In no case, however, are all gears simultaneously transmitting power, i.e., some are always running idly. In particular, on first speed only the first gear train is transmitting power; on second speed, the first and second trains; and on reverse, the first and reverse trains.

Automatic Control of Wilson Gearbox.—In order to simplify the method of operating the Wilson pre-selective gearbox so as to render it fully automatic it is arranged for the main spring of the gearbox to be replaced by a compressed air cylinder and the gear selection camshafts by electrical solenoids.

Automatic selection of the gear ratio according to the engine speed and throttle opening can be effected by the method devised by A. A. Miller, illustrated, schematically, in Fig. 182.

The device contains two electrical contact plates, namely, E, for changing-up and D for changing-down. The governor which is driven from the engine actuates a lever AFC, with hinge pin at F and an electrical contact at C. As shown the engine would be running slowly so that the contact C engages with the contact plate D and the low gear electrical circuit thus completed. The low gear would thus be engaged through an electromagnetic mechanism. If, however, the engine speed increases to a pre-determined maximum value the contact C would move upwards until it makes contact with the plate E.

The high gear engagement circuit would then be completed and a change to high gear effected. The plates D and E are cam-shaped and are carried on a sliding plunger B actuated by the rod P from the engine throttle control, which moves it to the left so as to raise both the low and high gear

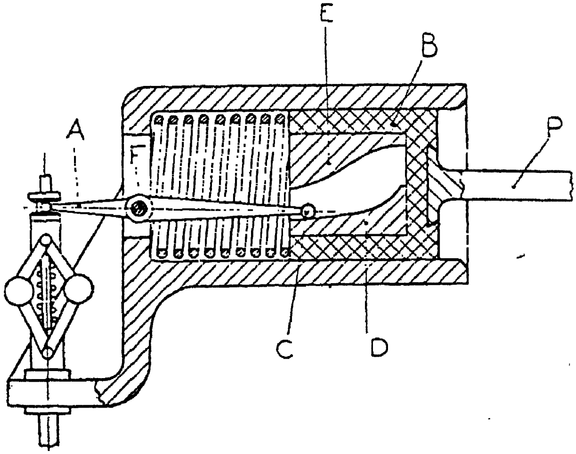


FIG. 182.—Automatic Control of Wilson Gearbox.

engagement speeds. By altering the cam shapes the circuits can be made for changing up or down at any desired speeds to suit the requirements of the engine and its transmission. This system which can be applied to the Cotal and other epicyclic gear systems, has been tested out with fully satisfactory results on an A.E.C. oil-engined motor bus.

Armstrong Siddeley Balanced Drive System. This arrangement enables the usual engine flywheel to be dispensed with by making the transmission system an integral part of the engine assembly ; the rotating members of the transmission then give the equivalent flywheel effect. Further, since the latter occurs closer to the propeller shaft improved running qualities are obtained.

The transmission in question comprises an automatic clutch of the centrifugal pattern described in Chapter IV and used principally for starting from rest and traffic driving purposes, a four speed pre-selector gear box and the usual universal joint and propeller shaft.

The engine and gear box unit are mounted flexibly on the chassis frame so that torque reaction and vibration effects are absorbed. The points of support, in this instance, have been carefully selected, after much experimental work, as it has been found that these locations have an important influence when considering the nature and directions of the forces due to torque reaction.

The Hudson Bendix Transmission System. A simplified method of gear changing which dispenses with the usual gear lever on the gear box and uses instead a miniature one mounted near the top of the steering column is used on the more recent Hudson cars.

Movements of this small gear lever in its 'gate' operates a selective form of electrical switch which energises one or other of a series of three solenoid magnets according to the lever position or 'gear selected'. The plungers of two of the solenoids operate pneumatic valves which control the movement of the piston of a relatively large vacuum cylinder. The piston rod is connected to the gearbox gear shift mechanism in such a way that the latter can be moved from one gear ratio to the next, so as to perform the normal operation of gear changing between top and 2nd.

The third solenoid operates another vacuum-control diaphragm for moving the gear shift mechanism sideways for obtaining the gear changes between 1st and reverse.

It will thus be seen that the vacuum chambers are used to supply the power for operating the gear shift mechanism of a normal type of gearbox, the selection of the gear being performed electrically by means of a selector switch and solenoid operated valves. In the earlier models the gearbox had the usual ball member, but with a hole left for the insertion of a spare gear lever, kept inside the car for hand operation if required.

The general layout of the Hudson-Bendix electro-vacuum gear change system is illustrated in Fig. 183*, the gear shift mechanism being shown in the neutral position.

Consider, first the operation of obtaining low gear by moving the small steering wheel mounted control lever to

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* *Automotive Industries.*

the position in its gate marked *'L.'* This movement across the gate establishes the primary contact between

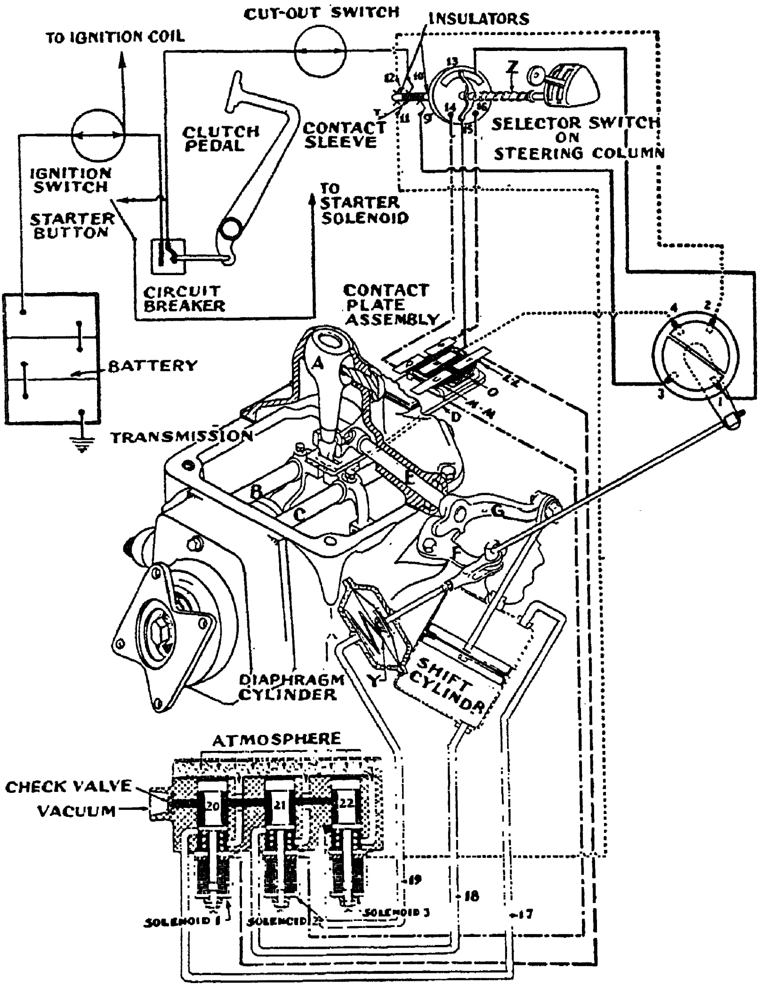


FIG. 183.—The Hudson-Bendix Transmission System.

main lead 10 and lead 11 (Fig. 183). Assuming the clutch to be depressed and the circuit breaker closed, this results in energizing solenoid 3, pulling down its valve and admitting vacuum to the diaphragm cylinder.

This moves the diaphragm back against spring *Y*, rotates bell-crank *F* clockwise, moves shaft *E* to the right and causes shifter *A* to engage into notch in shifter rod *C*.

Movement of the diaphragm has also caused a rotation of the interlock switch to the position shown in Fig. 184 (L. H. diagram). Movement of the finger forward into the 'low gear' position has connected contacts 13 and 16. Current now passes from main lead 10 through lead 12 through the interlock switch back to contact block 13 of the selector and out through 16 to plate *P* of the contact plate assembly.

From Fig. 183 it will be noted that this plate is connected through the sliding block *O* and its fingers, to plate *T*. Thus the circuit to solenoid 1 is completed, the latter is energized and pulls down its valve, admitting vacuum to the upper end of the shift cylinder, moving the piston upward.

This rotates crank *G* and rod *E*, resulting in forward motion of shifter *A* and shifter rod *C*, engaging the low gear. Link *D*, of course, moves with *A* so that as the shift is completed the contact fingers slide off plate *P*, breaking the circuit to solenoid 1. The valve of the latter under spring pressure returns upward, vacuum is broken and atmosphere admitted to the upper side of the shift cylinder piston. The latter is now balanced with atmospheric pressure on both sides.

Suppose we now shift the finger leaver into the second gear position. This connects main lead 10 with lead 9 and contact 13 with lead 14. If we now depress the clutch pedal, current starts through lead 9 to the interlock switch and (see Fig. 184 R. H. illustration) from there to plate *W*, through fingers *MM* to plate *U*, thereby energizing solenoid 2. Movement of the latter's valve connects the lower end of the shift cylinder to vacuum and starts the piston downward, rotating *G* and *E*, and moving *A*, shifter rod *C* and link *D* backward.

As the piston reaches the central position, fingers *MM* slide off plate *W*, breaking the circuit and de-energizing solenoid 2, thereby readmitting atmosphere to the lower end of the cylinder. The piston then stops momentarily. It will be remembered, however, that the diaphragm cylinder was left with the spring compressed, after the shift nto low was completed.

With the circuit to solenoid 3 broken, spring pressure tended to return crank *F* and rod *E* to the position shown in Fig. 183. Shifter *A*, however, could not move across the gate with gears engaged until the notches in shifter rods *C* and *B* were directly opposite each other.

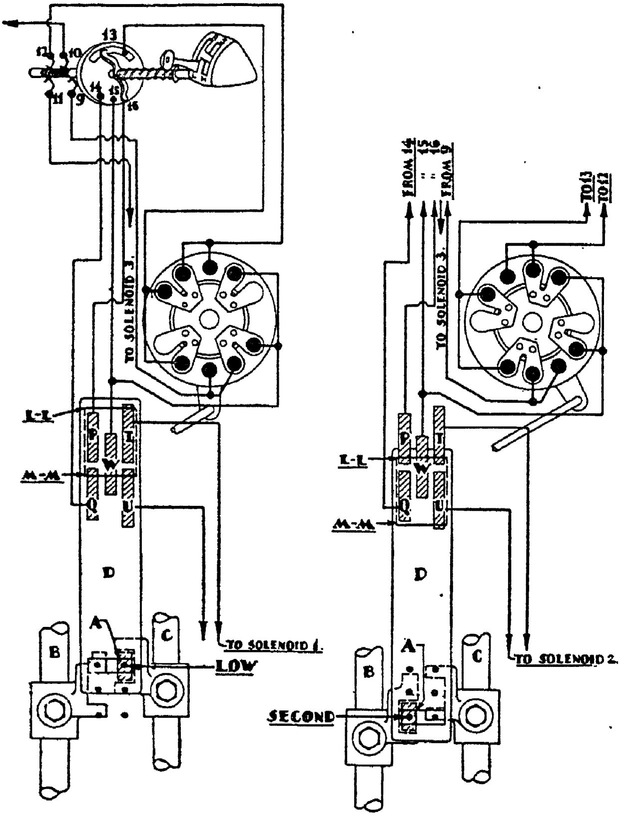


FIG. 184.—Showing Interlock Switch and Electrical Connections. (Left) Low Gear. (Right) Second Gear.

With the shift cylinder piston in its central position this has now occurred, and the force of spring *Y*, through the linkage mentioned, moves *A* across to engage the notch in shifter rod *B*, the transmission being in neutral.

At the same time, however, this movement of the diaphragm results in rotating the interlock switch to the

position shown in Fig. 184. This establishes a new circuit through the interlock switch to contact 13 in the selector

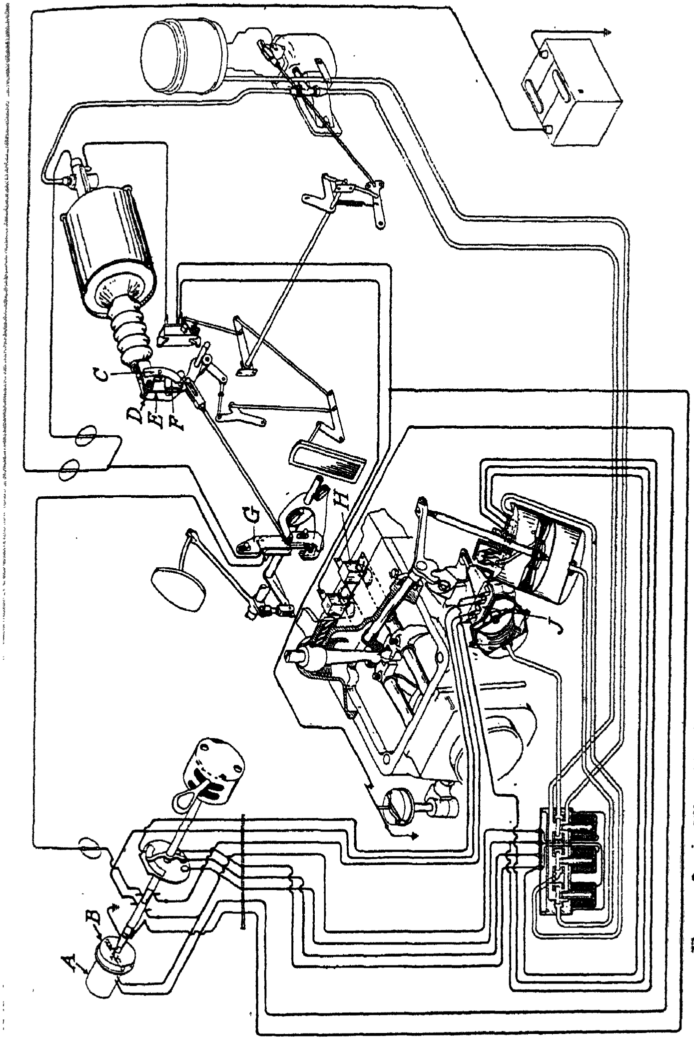


FIG. 185.—The Hudson Bendix 'Electric Hand,' showing later improvements.

A. Tooth abutment indicator solenoid
B. Detent rotor
C. Compensating lever trip
D. Valve lever
E. Bell crank
F. Compensating lever
G. Self-adjusting circuit breaker switch
H. Low-reverse rail switch

switch, and through lead 14 to plate *Q*, through fingers *MM* to plate *U* (see Fig. 183) to solenoid 2. Thus solenoid 2 is re-energized, vacuum is reconnected to the shift cylinder

and the piston moves downward until fingers *MM* slide off plate *Q* again, breaking the circuit with second speed engaged.

The method of changing from second to top gear is similar, whilst for the reverse gear the appropriate movement of the gear box striker is obtained by suitable switching from the control lever.

Certain refinements have been embodied in the later models of this 'electric hand,' including a tooth-abutment indicator to ensure complete engagement of the low-reverse gears before engagement of the clutch takes place. The general layout of the later design is illustrated in Fig. 185.

The Cotal Gearbox. This French transmission method, which has been used on private cars, racing cars and commercial vehicles, employs trains of epicyclic gears which are brought into action by clutches having iron faces operated by electromagnets. The gear box is light, compact, quiet in operation and easy to manipulate. It dispenses with the usual hand gear lever and in place employs a small lever and quadrant mounted on the steering column, similar in design and purpose to the pre-selective gear unit. In one design there are also numbers on the gear box itself, namely, 1, 2, 3 and 4 which are illuminated when the corresponding gear is engaged. The Cotal system also enables a 'coasting' position to be obtained.

Referring to Fig. 186* there are four electromagnets *A*, *B*, *C* and *D* of which *B* and *C* are fixed rigidly to the casing. The magnet *A*, driven by the input *N* carries the sun wheel of the first epicyclic train, while the magnet *D* is keyed to the output shaft *F* to which is also keyed the carrier *M* of the planet gears of the second epicyclic train. The outer member *H* of the first epicyclic train is made integral with a disc *G* interposed between the faces of the electro-magnets *A* and *B*. A similar disc *I* which forms the sun wheel *J* of the second epicyclic train is placed between the faces of the electromagnets *C* and *D*. The operation of the Cotal transmission system is as follows :—

* *Transmission Gear Developments.* L. J. Shorter. Proc. I.A.E Dec. 1937.

First Speed. The fixed magnets *B* and *C* are energized by a switch brought into action by the small selector gear lever on the steering column, causing the discs *G* and *I* with their gear elements *H* and *J* to be held stationary. In this manner a double reduction is obtained between the shafts *N* and *F*.

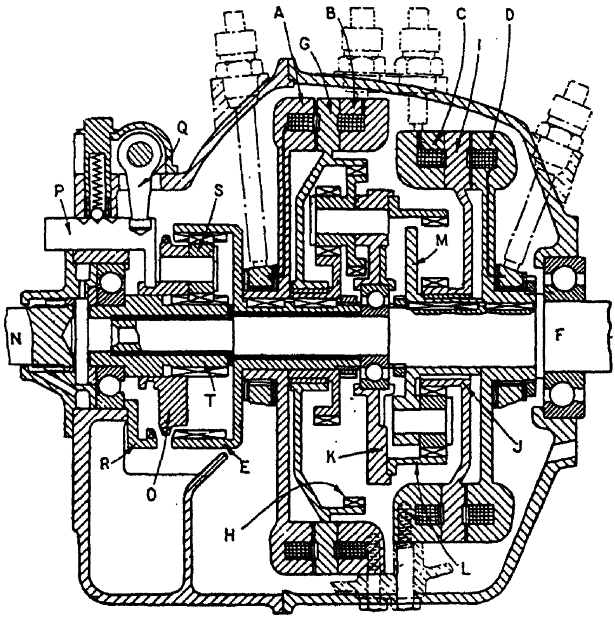


FIG. 186.—The Cotal Electromagnetic-Operated Epicyclic Gearbox.

Second Speed. By means of the selector gear and another switch the fixed magnet *B* and the revolving magnet *D* are energized. The first epicyclic gear train remains in gear and the second gear train is locked solid.

Third Speed. The moving magnet *A* and the fixed magnet *C* are energized so that the first epicyclic train is locked and the reduction is obtained on the second epicyclic train.

Fourth Speed. The selector lever switches cause both the moving magnets *A* and *D* to be energized thus locking both epicyclic trains solidly, giving a direct drive between shafts *N* and *F*.

Coasting. The selector lever is placed in neutral when the only part revolved by the propeller shaft is the output shaft *F* with its integral magnet disc and pinion carrier.

The current consumption needed for energizing the electromagnets is about 2·5 amps. (continuous); this is the same as that required for a headlamp bulb.

The efficiencies of the gears in their different gear ratio positions are relatively high. Test results obtained in the laboratory show efficiencies ranging from 99·5 per cent. for top or 4th gear (direct) down to 96·5 per cent for low or 1st gear.

It is now usual to employ a centrifugal clutch with the Cotal gear box; alternatively, it can be used with a hydraulic clutch drive.

The Maybach Pre-Selective Gearbox.—This gearbox, of German origin, is fitted to the Mercédès and has also been used on Lagonda cars.

It has five different forward speeds, so that the steps between each gear ratio are smaller and in consequence a higher average speed can be maintained on the lower gears than in the case of a three- or four-speed gearbox. The Maybach gearbox employs an automatically synchronizing dog-clutch operated by engine suction. The dogs of the clutches have sloping faces so that it is impossible for a pair of dogs to engage with each other until they are rotating at the same speed.

A vacuum cylinder, connected to the inlet manifold, is employed to compress or extend a double-acting spring, which imposes a load on the selector arm. While power is being transmitted, however, the dog-clutch will not disengage, as in addition to the end faces being machined at an angle, the driving faces are slightly undercut. As soon as the clutch is depressed, however, the dogs are disengaged by the spring pressure and the next pair engage at the instant the relative motion between them ceases.

The controls consist of an ordinary gear-lever, which operates a sliding wheel by hand, and a single lever below the steering column, moving about a vertical axis in a quadrant having four positions, which controls the vacuum valve, through which the three dog-clutches are operated.

The procedure in using the Maybach gearbox is as follows: To start away from rest, declutch and place the

selector lever in 'high first' gear. This is really a second speed gear, the lowest or 'low first' speed being obtained by the movement of the ordinary gear-lever.

When the car is moving in the 'high first' gear and is accelerating, the third speed is pre-selected by means of the handle under the steering-wheel, the accelerator released and the clutch disengaged. The gear then changes automatically and the process is repeated from any one gear to any other. To stop the car, the gear-lever is placed in 'neutral.'. The third position of this gear-lever gives 'reverse.' Gear-changing can be made both silently and easily with this device.

The Efficiency of Gears.—With well-designed gear teeth, and satisfactory lubrication it may be accepted that there is a loss of power of from 3 to 5 per cent. with each pair of gears engaged.

Miscellaneous Gearbox Items.—In order to avoid noisy gears, and to obtain maximum efficiency, the main shaft and layshaft should be as short, and as stiff as possible. The earlier gearboxes, with their long shafts, liable to 'whip,' were invariably noisy, and the gears wore more quickly. The layshaft should lie under the mainshaft, and should be about half immersed in gear lubricant. The use of engine lubricating oil is to be preferred, although some manufacturers recommend a thicker lubricant—possibly to quieten the gears. The highest efficiency, it has been shown by tests, is obtained with oil lubricant just sufficient in quantity to reach to the layshaft. An ample capacity filler plug accessible for the purpose, and a drain plug must be provided, in addition to the inspection cover.

The lubricant level should be readily observable, preferably by means of a dipstick gauge or filler-level elbow or plug in the side of the gearbox casing.

CHAPTER VI

ALTERNATIVE TRANSMISSION SYSTEMS

Although the method of transmission by positive gearing throughout, which has been described in the preceding pages, is now standard practice, there have been other interesting systems used. With the advent of mass-produced cars intended for use by all classes of drivers, the majority of whom have little mechanical knowledge or expert driving skill, it has become necessary to render the operation of gear-changing as simple as possible. It is for this reason that the car manufacturer has endeavoured, more recently, to dispense with the 'crash change' type of gearbox which, as we have seen, requires a certain amount of skill and much experience to produce noiseless gear changes up and down.

Instead, it is now usual to find the later model cars fitted with gear change devices, e.g., free-wheels, synchromesh and pre-selector gears which dispense with the earlier gear changing difficulties and, incidentally, prolong the lives of the gears themselves by obviating gear clashing when changing gears. There can be little doubt, to those who have followed motor developments from the earliest times, that the ultimate transmission for the reciprocating type of petrol engine will be progressively variable instead of in three or four steps, and automatic in action; that is to say, the gear ratio will be adjusted automatically to suit the resistance encountered—for example, the gradient. Already there are signs of important developments in this direction.

It is not surprising, therefore, to learn that from the earliest times attempts have been made to invent and construct alternative systems of drive from engine to wheels, which did not entail so much trouble on the part of the driver in their operation. Of these systems, some were mechanical in principle, others pneumatic or hydraulic, and some electrical. We shall refer, briefly, to some typical examples of these transmission systems, some of

which are now of historical interest only, mentioning, also at rather greater length, the more recent forms. Each of these systems has for its object the variation of the torque or driving effort (and therefore the speed-ratio) of the rear wheels relatively to the engine speed.

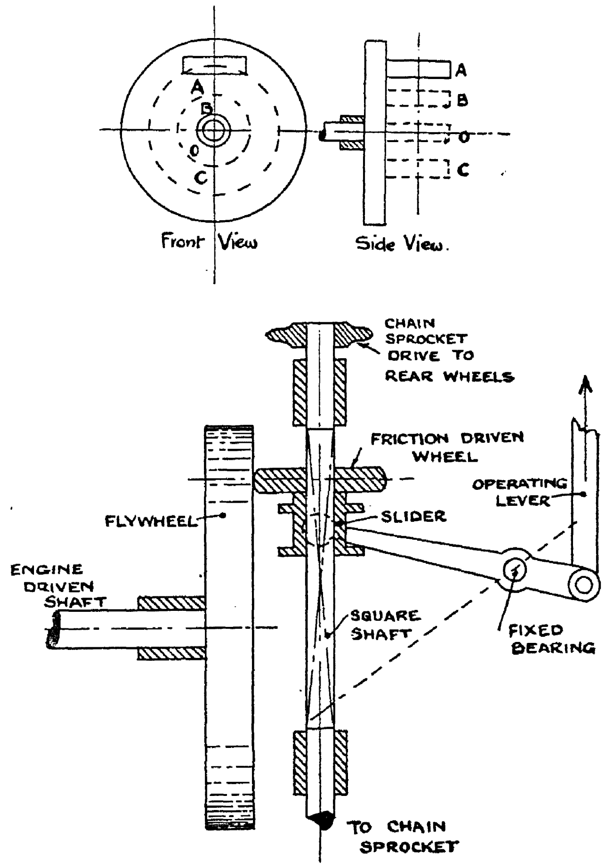


FIG. 187.—Principle of Friction Transmission (G.W.K.)

Friction Drives.—An attractive mechanism, long known to engineers as a method of varying the speed of one driven member relatively to that of the driving member, gradually, and over a wide range, including the reversal of the direction of motion of the driven shaft, employs

the principle of a friction-material faced disc driven by a friction wheel. Fig. 187 illustrates in outline this method. Referring to the upper diagram, a flat circular disc is driven through suitable couplings from the engine. Another wheel, *A*, which is usually constructed of friction material such as fibre, or fabric, bolted between two steel discs, is arranged in frictional contact with it, so that as the latter rotates, it causes *A* to rotate. If there is no slip, both wheels will rotate at the same linear or peripheral speed. By making the radius of the wheel *A* equal to the radius *OA*, the speeds of both wheels will be equal. Next, suppose that *A* is slid towards the centre, to the position *B*, such that the radius *OB* is one-half of the radius *OA*. It is evident that the fly-wheel must make one-half a revolution in order to turn *A* through one complete revolution. This gives a speed reduction of one-half to *A*. Again, suppose that *A* is slid to the central position *O*, it will not rotate, since the point *O* has no peripheral speed. Finally, imagine that *A* is moved past *O* to the position *C*. It will be obvious that the direction of rotation of *A* will now be reversed, but the speed will be equal to that corresponding to the position *B*.

We have thus a system giving a gradual variation from engine speed down to zero, and through zero to reverse.

In practice, the fly-wheel driving disc was made of cast-iron, and formed either the fly-wheel of the engine, or was connected by means of an intermediate shaft, with flexible joints, to the engine shaft. The friction disc may be made from fabric, fibre or cork, held rigidly between two circular steel plates. The boss of the friction wheel was provided with a collar which was engaged with the end of the gear operating lever (Fig. 187). The friction wheel slid on a squared or castellated shaft, to one end of which was usually fixed a chain sprocket, to transmit the drive to the rear axle sprocket; the latter was of larger diameter, so that the usual final drive gear reduction was obtained. In drives of the type described, it was not usual to provide a differential gear in the rear axle, although it was merely a matter of suitable design to incorporate this. There need be no clutch, for the driven wheel is arranged to move opposite

to a central recess, where it is out of contact. Actually, however, a clutch is an advantage in gear changing.

In practice it was usual to provide four or five 'forward' positions for the gear operating lever, by means of notches in its quadrant instead of giving an infinite number of positions.

In one example, it was arranged to increase the pressure between the two surfaces in the low gear and reverse positions, but this calls for careful design; usually the pressure variation is obtained by means of springs which give a greater pressure in the lower gear positions.

If the width of the friction disc A is appreciable, it will be evident that the outer edges will tend to rotate faster, and the inner edges slower than the mean peripheral circle of the flywheel. The material will, therefore, be subject to tension and compression in these places, *i.e.*, slip will occur.

It will be obvious that since the torque of the driving disc is constant, that of the driven wheel A will increase from circumference to centre of the flywheel disc, *i.e.*, from C to O, so as to provide the increasing torque at the lower speeds.

The friction type of variable speed drive has several modern applications in engineering machine practice and there are some eminently successful variable speed units in everyday use in connection with machine drives. Most of these units depend upon frictional principles, tapered cone pulleys or expanding vee-pulleys with belts; from the automobile application viewpoint none appears to be so efficient, positive in action and enduring as the gear transmissions previously described.

Chain Drive.—This method of driving the rear wheels which was once common to motor-cars, three-wheelers and commercial vehicles. It is a very efficient system—probably of higher efficiency than the bevel wheel final drive when properly lubricated and protected—but becomes noisy as chain stretch and sprocket wear occurs.

Fig. 188 illustrates a typical arrangement for final chain drive. There is the usual bevel-drive and differential gear casing driving two axle shafts, to the outer ends of which the chain sprocket wheels are secured. The rear wheels with their sprockets attached run freely on the plain, or

'dead' axle shown; the latter is merely a distance piece to keep the wheels in their proper positions.

This rear axle which is attached to the frame by the usual rear springing system is connected to the front casing by two radius rods, each of which has a journal bearing on the former axle and a hinged joint (or universal) where it is attached to the front casing. These radius rods transmit the thrust of the road wheels to the chassis frame, and allow the wheels to move in an arc (under the influence of the springing action) about the front sprocket centres, or axis, so that the chains do not tend to slacken or to tighten as the rear axle and wheels move up and down. Further, each radius rod is made in two portions, screwed

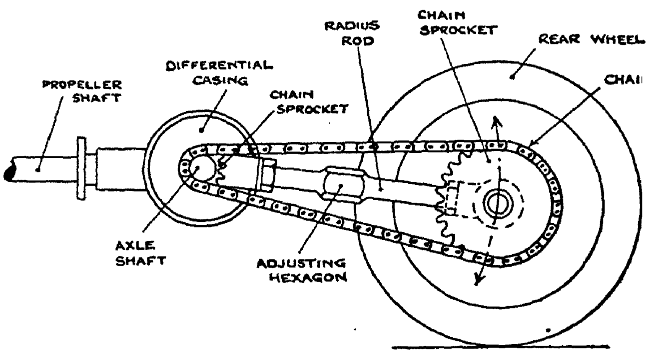


FIG. 188.—The Final Chain Drive System.

together in such a way that the length may be altered; this enables the tension of the chains to be adjusted, conveniently, and the rear axle to be aligned with the front ones. Brake drums can be fitted to each, or both sprocket shafts or hubs. The springs are shackled at both ends, as a rule.

Chain drives, as we have stated, are apt to be noisy; the substitution of silent type roller chains to some extent alleviates this drawback. It is usually difficult to keep chains clean when in use, due to mud and water ingress, unless chain casings are employed.

Hydraulic Transmission.—Although not yet used upon modern cars, except for the 'fluid flywheel' hydraulic

clutch, this type of transmission which has several attractive features has been applied experimentally to cars in the past. It has also been successfully used for marine and locomotive transmissions and, more recently to commercial motor vehicles.

In this system, the clutch, gear-box and rear axle drive are dispensed with, and in place is provided a pump, or series of small pumps embodied in a casing attached to the engine, the pumps being engine driven. These pump a fluid—usually oil, on account of its low frictional losses—to two hydraulic engine systems, one on each rear wheel.

It is, of course, well known that oil under pressure can be made to work an oil, or hydraulic motor in the same way as compressed air, or steam, can work a suitable engine to give power. The rear wheels of the car can, therefore, be driven by means of the oil-motors, from oil supplied under pressure from the engine-driven pumps. To obtain 'neutral' gear, it is only necessary to 'byepass' the oil through suitable valves so that no pressure is transmitted to the wheel motors. Gear ratio variation is obtained in the well-known Lentz system by employing two sets of hydraulic motors, one on each rear axle shaft, and three rotary sliding vane pumps of different capacities in connection with these. By varying the combinations of these pumps to the two motors, the speeds can be varied. Thus, if one pump only is supplying both motors a high gear ratio is obtained; if all three pumps are in operation, then a low gear ratio is given. The engine sets, it should be explained, consist of two units of different capacities, the larger units having a common supply connection from the three pumps, and the smaller units also. This arrangement enables an hydraulic differential action to be obtained.

In other hydraulic systems, speed variation is obtained by varying the strokes of the engine and motor pumps, either by means of a swash-plate or eccentric cam of variable throw. The best known hydraulic gear types are the Hele-Shaw, Lentz, Vickers-Coats, Lysholm-Smith and Salerni systems.

In two or three of the more recent systems intended for automobiles, the pump used is a centrifugal one delivering oil to a single or multi-stage turbine—usually with

hinged guide vanes operating automatically. As applied to motor vehicles it is usually necessary to provide a free-wheel device to relieve the converter from over-run effects, as when the vehicle is coasting down inclines. A mechanical type of reverse gear must also be provided unless the hydraulic equivalent is embodied in the converter.

It is generally desirable also, when operating under conditions equivalent to the top-gear ratio, to employ a direct-drive from the engine to the propeller shaft in order to cut out the hydraulic converter under these conditions. In the Leyland converter a special clutch-coupling is provided for this purpose.

The Leyland Torque Converter.—The hydraulic torque converter used upon the Leyland commercial vehicles operates upon the Lysholm-Smith system, whereby the

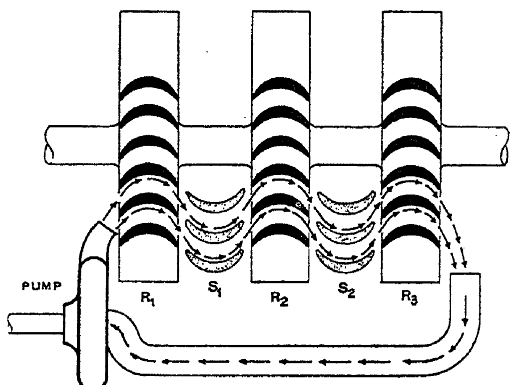


FIG. 189.—Principle of Lysholm-Smith Torque Converter.

gear ratio is changed automatically and the usual clutch unit is dispensed with; the only control is therefore the engine accelerator pedal. With this system the vehicle can be started from rest with the engine running, merely by opening the engine throttle; when the foot is taken off the accelerator pedal and the brakes applied the vehicle comes to rest with the engine 'idling'. Acceleration and hill climbing operations are carried out with the accelerator pedal, the performance being superior to that obtained with the ordinary clutch and gear-box.

The principle of the torque converter is illustrated, diagrammatically in Fig. 189. The centrifugal pump, which is driven directly from the engine delivers oil to the blades of three turbine rings R_1, R_2 and R_3, between which are arranged two sets of stationary blades S_1 and S_2 fixed to the casing. The path of the oil is indicated in the diagram by the arrows. After leaving the third set of blades R_3 the oil returns to the pump. Owing to the shapes of the blading and to the fact that the fluid impinges upon three sets of blades coupled in series the torque can be increased up to the ratio of 4·8 : 1. Labyrinth seals are provided to prevent leakage of the oil between the rotor and the casing; a slight leakage, however, is allowed for lubrication purposes.

In addition to the hydraulic unit described there is a double-acting clutch for connecting the engine either to the propeller shaft and back-axle or to the converter as required. It should be emphasized that this member is not a clutch of the usual automobile type but is merely a coupling. When connected to the converter with the engine idling the vehicle remains stationary until the accelerator pedal is depressed.

When connected to the propeller shaft it gives a straight through drive or 'direct' drive thus cutting out the converter unit. The Leyland torque converter unit also embodies a free-wheel device to obviate over-running of the converter under certain operating conditions.

Fig. 190* shows the complete converter unit in sectional view. Referring to the clutch member shown on the left the clutch disc A (nearest the engine) is connected to a shaft passing through the centre of the converter to the propeller shaft and rear axle; the engagement of this clutch member connects the engine with the rear axle, and gives 'direct' drive under these conditions.

The second clutch disc B is mounted on a hollow shaft C; the other end of this shaft is connected to the impeller of the centrifugal pump inside the converter casing; this forms the drive to the converter. It should again be emphasized that as the clutches act as couplings only, they are not subjected to any appreciable wear.

The driven plate D is provided with an intermediate

* Courtesy of *Machinery*.

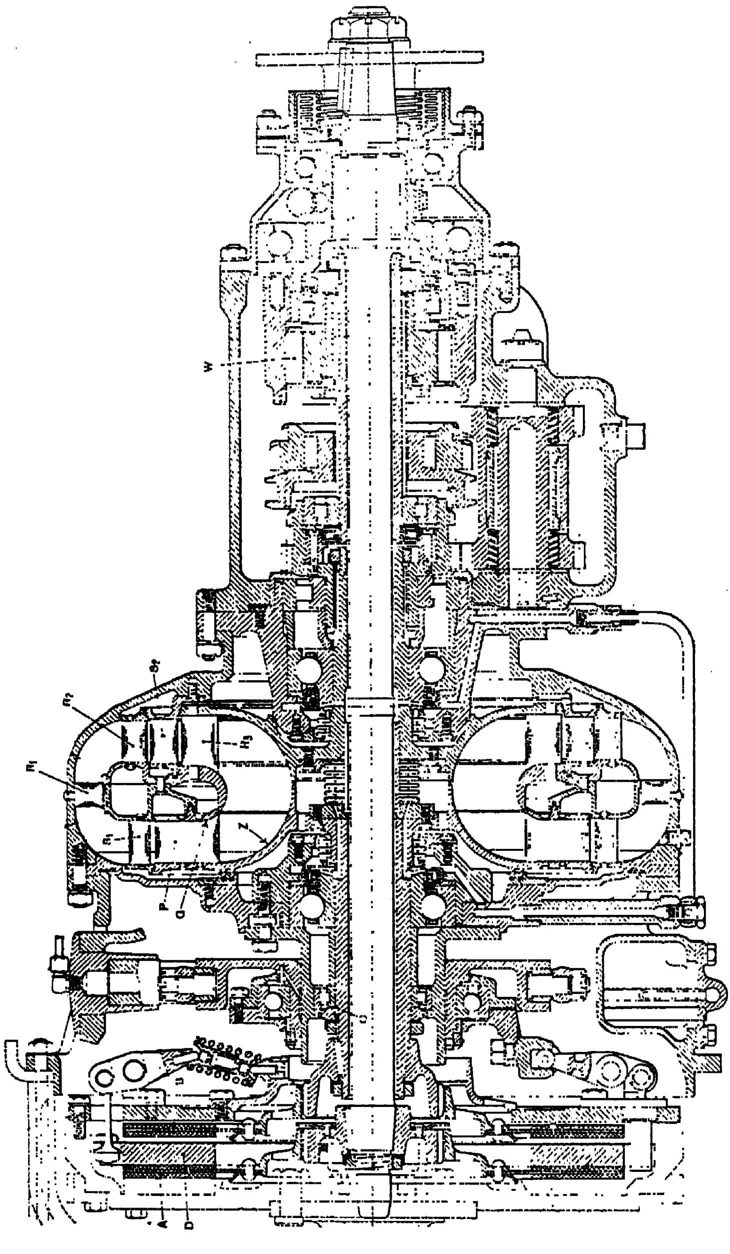


FIG. 190.—The Leyland Torque Converter Unit.

position so that both clutches can be disengaged; this is necessary for starting the engine.

In reference to the lettering for the turbine and guide blades, shown in Fig. 190 this is identical with that given in Fig. 189. In regard to other letters the pump blades are shown at *P*; the centre carrier ring at *G*; the pump centre at *Z* and the free-wheel member at *W*. The latter is provided on the shaft driven by the turbine rotor so that the converter is completely isolated and comes to rest immediately its clutch is disengaged; this permits a direct drive in which no hydraulic losses occur. As previously mentioned it also renders coasting possible with safety when the vehicle is being driven through the converter. The free-wheel has a number of rollers positioned in recesses around the inner or driving member in order to lock the forward motion. The inner member is driven by the converter and the outer one is coupled to the propeller shaft. The reverse gear is located with the free-wheel in a kind of gearbox which is bolted to the rear end of the turbine casing (Fig. 190); it is of the conventional layshaft type and is operated by means of a hand lever in the driver's cab.

The oil system of the converter includes an oil cooler through which the oil circulates when the converter is in action.

The fluid used consists of a mixture of lubricating oil and paraffin.

The Bendix Turbo-Flywheel Transmission. A combination of an hydraulic coupling and an automatic gear change of the planetary type. Referring to Fig. 191* the hydraulic coupling unit has an impeller *I*, first rotor R_1, reaction member *S* and second rotor R_2. The impeller *I* is secured to the flywheel whilst the rotors are both secured to the hollow driven shaft and the reaction member *S* is mounted on a stationary hub extended inward from the rear part of the flywheel housing so that it is free to rotate in one direction only. The whole mechanism is virtually carried on a central shaft extending through the transmission. In the forward drive operation this shaft is coupled to the engine crankshaft by means of a

* *Automotive Industries.*

toothed clutch. At the start about 50 per cent. of the engine power passes from the flywheel to the sun gear A of the planetary system through this central shaft.

The action of this transmission system is, briefly as follows :—

In forward drive the sun pinion A is connected to the crankshaft by the clutch teeth on the central shaft ; ring gear B is connected to the hub of rotor R_1R_2 by a toothed clutch between the hydraulic unit and the planetary

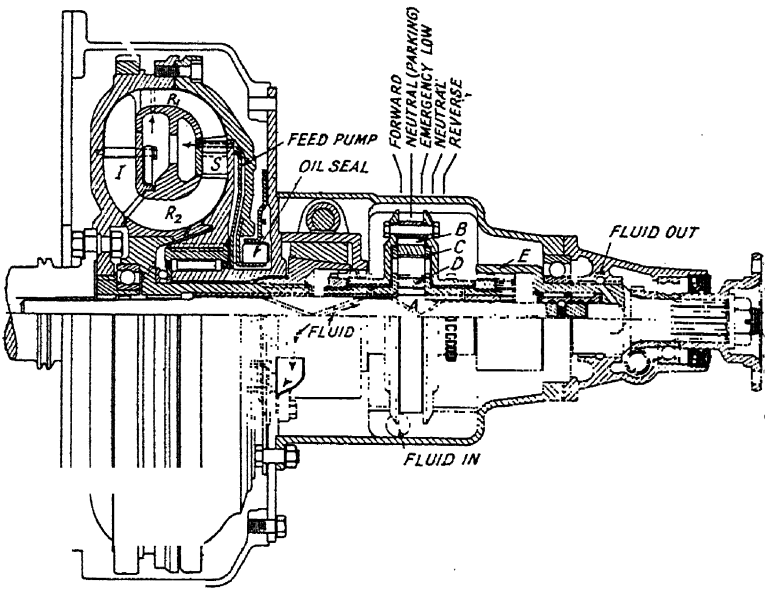


FIG. 191.—The Bendix Turbo-Flywheel Transmission System.

assembly, and planetary carrier C is connected to tail shaft E by a toothed clutch within the cup-shaped inner end of the shaft. Both the sun pinion and the ring gear turn in the same direction, and they carry the planetaries and carrier C along with them. When the drive is ' direct,' the sun pinion and ring gear turn at substantially the same speed, and they then contribute to the power transmitted in proportion to their pitchline velocities, which are proportional to their pitch diameters (1 : 3 in Fig. 191). At the start, of course, there will be considerable slippage in the hydraulic device ; the ring gear then turns at lower speed than the

sun pinion, and the proportion of the power transmitted by the hydraulic unit is them relatively less.

In 'emergency low' ring gear B is locked to the housing by the toothed clutch between the hydraulic unit and the planetary assembly, and therefore remains stationary; planetary carrier C remains connected to tail shaft E and sun pinion A is connected to the hub of the hydraulic rotor by the clutch teeth on the central shaft. Thus the power is first transmitted through the hydraulic unit and then through the planetary assembly, which latter, with the sun pinion as the driving and the planetary carrier as the driven member gives a reduction ratio of 4 : 1.

In 'reverse' the sun pinion remains connected to the hydraulic rotor, the planetary carrier is now locked to the housing by the toothed clutch between the hydraulic unit and the planetary assembly, and the ring gear is clutched to the tail shaft. It is evident that in this case, with the planetary carrier stationary, the ring gear turns in opposition to the sun pinion and the tail shaft in opposition to the crankshaft, hence the direction of the car is reversed. The reduction ratio (in Fig. 191) is now 3 : 1.

Although there are four positions for the shift lever, no gear changing is done after the car has been started, all subsequent changes in speed ratio or torque ratio taking place automatically. The four positions of the shift lever correspond to the following operating conditions:—

Position	Direction	Combination of elements and torque ratio.
1.	Reverse	Hydraulic turbo. Mechanical reverse. Variable ratio.
2.	Neutral	Engine may be moved around freely.
3.	Forward	All ratios down to 1 : 1.
4.	Forward	Emergency low. Ratio variable.

A positive direct drive can be furnished with the gear as well as for using the hydraulic unit as a brake. Only a single train of gears is needed to perform the various gear functions. The emergency low-speed ratio is obtained by hydro-mechanical means without the addition of extra gears, and lower speed ratios are available for descending

severe grades. A hydro-mechanical reverse is obtained with the same gears that are used for the emergency low speed. By means of the hydro-mechanical emergency low gear the car may be made to climb over a curb or out of a deep rut, or ascend a grade in excess of 50 per cent. with ease, provided there is sufficient traction.

The Miller Semi-Automatic Transmission. This American transmission system employs an hydraulic coupling unit *A* (Fig. 192); a free-wheel or over-run clutch

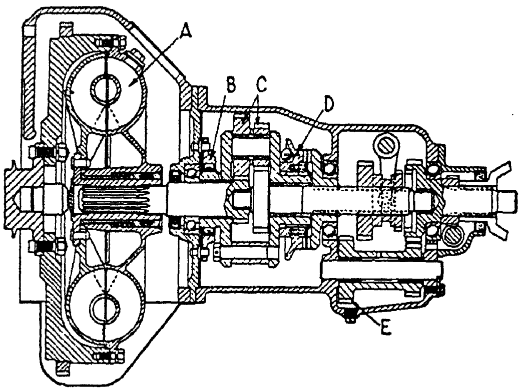


FIG. 192.—The Miller Semi-Automatic Transmission.

B; a compound spur reduction gear *C*; a dog clutch *D* engaged by centrifugal force through a number of steel balls and a reverse gear *E*.

When the car is started from rest, power is transmitted through the reduction gears, the reaction on the carrier of what might be called the back gears being taken by the over-running clutch *B*. When such a speed has been reached that the driver wishes to change to the direct drive, he releases the accelerator pedal for a short time; the reaction due to the reduction gears then ceases and the gear carrier with the gears starts rotating in the forward direction with the speed of the propeller shaft. This energizes the steel balls, which move radially outward, thereby engaging the jaw clutch *D*, which puts the car in direct drive.

A certain minimum speed is required before the shift into direct drive can be made, but the driver may accelerate

the car in low gear to a much higher speed if he desires. Shifting to low is accomplished in the same manner whenever the car speed is below the mininum at which the direct drive clutch can be engaged.

It is stated that a gear reduction of about 2 to 1 in conjunction with the hydraulic coupling will give better acceleration than the usual three speed gearbox arrangement.

The Hydra-Matic Transmission. This automatic transmission which was introduced on the 1940 model Oldsmobile cars is a development of the semi-automatic transmission used on the earlier model cars.

It employs a fluid flywheel coupling and four transmission speeds, with a reverse also. There is no clutch pedal to operate and the gears move into action automatically through the four speeds as the car accelerates. In starting from rest a starter button is depressed with the foot and a lever on the steering column is set into 'forward' or 'reverse' position as required and the accelerator depressed. Once the car has been started it can be driven at any speed, stopped and started again without any other controls than the accelerator and foot brake. Even on steep gradients the engine cannot be stalled.

The hydraulic coupling unit is similar in principle to the 'fluid flywheel' previously described. It has two rotating members, one of which is attached to the flywheel and the other to the transmission. There are 48 radial vanes and the operating faces of the two members are $\frac{1}{32}$ in. apart.

The Cadillac Hydra-Matic transmission employs planetary or epicyclic gear units combined with the fluid coupling member and it employs a low-reduction rear axle. The intermediate planetary unit is a two-step one giving a reduction ratio of 2·26 : 1.

The manual control consists of a small lever and indica or mounted on top of the steering column beneath the steering wheel. This indicator shows four positions—'Neutral,' 'Hi,' 'Lo' and 'Reverse'—and it is illuminated for night driving. For all normal driving the 'Hi' position is used. When the car is started in this position, the transmission shifts automatically through the four speeds.

When set in 'Lo,' third and fourth speeds are locked out, so that the transmission remains in either first or second gear, making it possible to use the engine as a brake when descending hills.

To start a Cadillac from rest with Hydra-Matic transmission, the operator starts the engine with the gear selector in neutral, selects the direction by placing the selector level in either 'Reverse' or 'Hi' position, and then presses the accelerator.

The advantages claimed for this transmission are that it is completely automatic, eliminating the clutch pedal and gear-shift lever and the need for their operation; that it eliminates jerky starts, quietens the engine by lowering its speed of operation, and damps out vibration and shock. Acceleration and hill-climbing ability are increased, and the car is said to have better traction on slippery surfaces. The automatic transmission, moreover, is claimed to reduce fuel and oil consumption, and wear and tear on the engine.

The components of the transmission (Fig. 193), as described in *Automotive Industries*, consist of a flywheel A bolted to the engine crankshaft, while fluid coupling cover B is bolted to the flywheel and is splined at C to the front-unit drive-gear D. For the first gear, sun gear E of the front unit is held from rotation by brake band F applied to drum G. Planet-carrier assembly H, which now rotates at 0.694 times crankshaft speed, is keyed to the intermediate shaft I, which in turn is splined to the impeller of the fluid coupling. The runner of the fluid coupling is splined to main drive shaft J. From the main drive shaft the power passes on through the rear unit, whose ring gear, K, is held from rotation by brake band L acting on drum M, sun gear N causing the rear-unit planet-carrier assembly O to rotate at 0.395 of the speed of main drive shaft J. Rear unit planet-carrier O is integral with output shaft P, which rotates at $0.694 \times 0.395 = 0.274$ times crankshaft speed.

The brake band of the front unit is applied by hydraulic pressure, while the brake band of the rear unit is applied by both spring and hydraulic pressure. The fluid coupling is kept filled under pressure by an oil pump.

In second gear, the front unit is locked by clutch Q, so that cover B is coupled to the impeller of the fluid

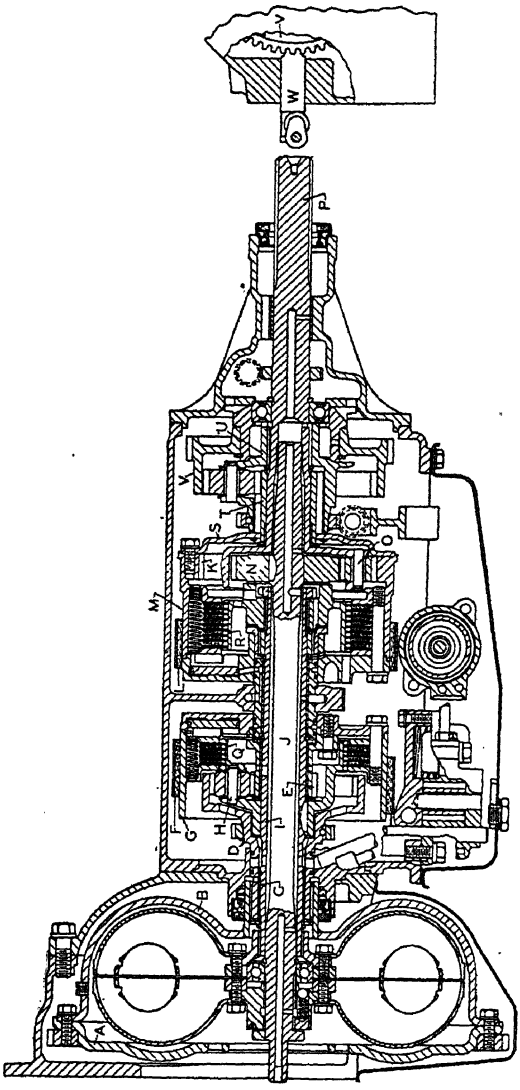


FIG. 193.—The Oldsmobile Hydra-Matic Transmission

coupling, and only the rear planetary unit is active, which, as already mentioned, gives a reduction to 0.395 times crankshaft speed. When in second gear, the front-unit brake band is released by hydraulic pressure, while the

front-unit disc clutch is engaged by hydraulic pressure.

In third gear, the front planetary unit is active again, the same as in first gear. When the power enters intermediate shaft I the torque is divided, approximately 40 per cent. passing to the fluid coupling, the remaining 60 per cent. from the front unit rearward through intermediate shaft I to disc clutch R and ring gear K of the rear planetary unit. The portion of the torque transmitted to the fluid coupling is applied to sun gear N of the rear unit. With torque in the direction of crankshaft rotation applied to both sun gear N and ring gear K, planet-carrier assembly O and output shaft P are carried around with them—at 0.694 times crankshaft speed (neglecting slippage in the fluid coupling). The rear-unit brake band is released by hydraulic pressure overcoming spring pressure, while the rear-unit clutch is operated hydraulically.

In fourth gear both planetary units are locked, and the drive is then direct, except for the slippage in the fluid coupling.

In reverse, the flow of power is the same as in first gear forward up to sun gear N of the rear planetary unit. Here the torque divides, part being applied to rear-unit planet-carrier O, the remainder to rear-unit ring gear K, which latter is connected through drum end S to sun gear T of the reversing planetary gear. That portion of the torque which is applied to rear-unit planet-carrier O (which is in driving connection with the output shaft) is combined with the remainder of the torque, which passes from rear-unit ring-gear K through reversing-unit sun gear T to reversing-unit planetary carrier U and thence to output shaft P. In this case the rear unit does not act as a planetary gear in the ordinary sense, as all three of its members are rotating. Sun gear N is the driver, while ring gear K and planet-carrier O are the driven members. The torque impressed upon planet-carrier O is transmitted directly to output shaft P, while that impressed on ring gear K is carried through drum end S to sun gear T of the reversing unit, and after being modified in that unit, is transmitted through the reverse-unit planet-carrier U to output shaft P, to which the planet-carrier is splined. In reversing, the rear-unit brake band is released hydraulically and reversing "pawl" W is engaged manually with

external gear teeth on ring gear V of the reversing unit, through suitable linkage.

Electrical Methods.—In the past, motor vehicles have been fitted with a combination of petrol-engine and electric (or magnetic) transmission systems. The general principle employed is to dispense with the gearbox and clutch of the ordinary motor-car, and in place to arrange for the engine to drive a dynamo, the electric current from which is supplied to a single electric motor, driving through the usual propeller shaft, the rear axles, or to a pair of motors, one on each rear axle shaft; usually the former system is employed. Variation of speed, and torque, is obtained by varying the strengths of the currents in the field and armature circuits of the motor or motors. Since the petrol engine works most economically at a certain speed, these systems aim at keeping the engine running at its most economical speed always. Examples of successful petrol-electric types of commercial vehicle which have been used in this country include the Tilling Stevens, Thomas, and Magnetic cars.

The Electric Transmission System.—This system was successfully used in the case of motor-omnibuses and commercial vehicles, in both this, and other countries, but it is now widely used for rail-cars, streamline diesel trains and ship propulsion, in combination with petrol

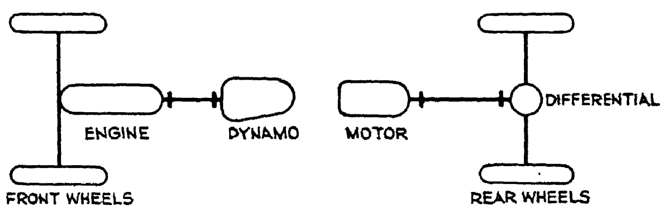


FIG. 194.—Principle of Electrical Transmission.

or diesel engines and steam turbines, respectively. In one case an ordinary four-cylinder petrol engine is employed, and its power is transmitted through a spring drive to an electric generator, or dynamo, which

transmits its electrical energy to an electric motor placed in the usual gear-box position, and connected to the usual propeller shaft form of drive and to the rear axles. There is no mechanical connection between the engine and the back axle, no gear-box and no clutch. The output of the dynamo is controlled by varying the speed of the engine by the usual throttle accelerator method, in conjunction with variable resistances which regulate the field strengths of dynamo and motor. The regulation is effected in a very simple manner, by operating a control lever on the steering column.

In a typical instance the current from the dynamo is taken by means of cables through the control switch mentioned; the latter usually has three positions, namely, forward, neutral and reverse. The driving of this type of vehicle and its control is very simple. The throttle accelerator pedal and the control lever are the only two items concerned. *The vehicle is started* by releasing the hand-brake—the engine just 'ticking' over—placing the control lever in the 'forward' position, and depressing the accelerator pedal. At the lowest, or idling speed of the engine there is not sufficient current generated to provide enough energy to move the vehicle. As soon as the engine is speeded up, however, the dynamo delivers sufficient current to the motor to give it the necessary starting and accelerating torque. The starting from rest process is very smooth, due to the progressive response of the engine to the throttle and the consequent gradual energising of the magnetic field of the electric motor. No sudden depression of the accelerator pedal can, therefore, cause a jerky start, as with the ordinary clutch. *To stop the vehicle,* it is only necessary to release the accelerator and to apply the hand-brake.

This gradual and continuous transmission of power from the engine to the driving wheels, saves the engine and the transmission (*i.e.*, back axle and propeller shaft) from the usual driving shocks associated with mechanical transmission systems.

Compared with the 'direct drive on top gear' mechanical transmission, the dynamo-motor drive system described cannot be so efficient, for there is a double loss of efficiency in the latter case. Thus, if each electrical unit has an

efficiency of, say, 85 per cent., the overall, or net efficiency of the two is $85 \times 85 = 72 \cdot 25$ per cent., which is appreciably lower than the 85 to 90 per cent. for the direct mechanical drive. On the other hand there is an overall saving in the starting operations, freedom from transmission shocks, a higher top gear ratio on the level or down hill, insulation of the engine from the rear wheels, better starting on steep hills, and increased safety, for the vehicle without having the brakes applied, on a steep hill, will only crawl backwards. The electric generator may be used for stationary power generation purposes.

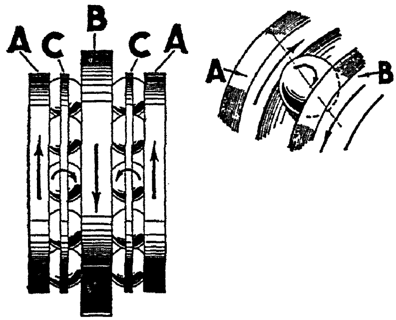
Variable Transmissions: Mechanical.—Apart from the frictional, electrical and hydraulic transmission systems described, attempts have frequently been made to provide a purely mechanical transmission which shall give a variable gear ratio from top forward speed right down to neutral, and thence to reverse speed. A large number of patents have been granted for such devices, but only one or two have found any practical application. Many of the ideas put forward have involved the use of expanding and contracting pulleys, gear, and chain wheels, epicyclic systems and variable cams; difficulties of manufacture, expense, excessive wear and noise, and excessive weight have been the chief drawbacks of these devices. There are, however, one or two promising systems to which a brief reference will be given here.

The Hayes Automatic Transmission.—This automatic variable speed transmission, which has been fitted to certain of the larger models of Austin cars, has no external control—other than the usual clutch for starting and stopping, and a lever fitted for selecting the forward and reverse positions; the forward gear ratios are altered automatically to suit the engine speed and total resistance experienced by the car on the road.

The principle of the transmission is illustrated in Fig. 195 (A), which depicts a double ball thrust race. Normally the central-race *B* is kept fixed, whilst the races *A* rotate in the same directions; the ball-race cages *C* also rotate.

If, however, the central race is left free and the cages *C* are fixed, then the two outer races will, as before, revolve in the same direction, and the balls will be rotated in their

fixed cages *C*, thus imparting, by frictional action, a rotating effect to the race *B*; the latter will therefore rotate in the reverse direction to the races *A*. Fig. 195 (B



(A) (B)

FIG. 195.—Principle of Hayes Transmission.

shows one of the balls between two races *A* and *B*, the arrows indicating the directions of movement of the outer and inner races as well as that of the ball in the case of a journal type ball bearing having the cage fixed.

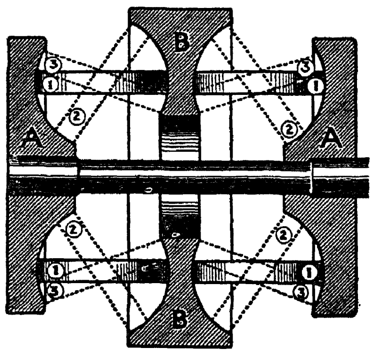


FIG. 196.—The Hayes Variable Transmission.

In the case of the Hayes transmission Fig. 196 the same principle is employed, but instead of having flat races and balls, as in Fig. 195 (A) specially contoured races *A* and *B* are employed, in conjunction with narrow rollers 1. The curvature of the two outer races has been increased so as to give 'toroidal' surface.

With the rollers in position 1, able to rotate on their own axes, yet unable to travel around the race, as they are anchored to an outside object, the transmission from the two outer races *A* to the double-centre race *B* will be exactly the same as in the case of the double ball thrust bearing shown in Fig. 195 (A).

As the rollers, when in position 1 rotate on tracks of the same diameter on both *A* and *B*, the speeds of *A* and *B* will be the same, although in opposite directions. If, however, the rollers are tilted to occupy position 2, they will be driven by a smaller diameter of *A* on to a larger diameter of *B*. If the diameter of contact of *A* is one-third that of *B* the race *B* will revolve at one-third of the speed of *A*; the gear ratio will then be 3 : 1. This tilting of the rollers is the means employed for varying the gear-ratio. It is merely a matter of rocking or precessing the rollers to engage with the races *A* and *B* on different diameters.

Given a gradual form of control, an infinite variation of ratio is thus rendered possible between the limits of the angular movement of the rollers. This, as Fig. 196 shows, is very considerable, ranging from a low gear of approximately 4 to 1, to an over gear (position 3) of 1 to 1·7, the engine then rotating once for ever 1·7 revolutions of the propeller shaft. Apart from the ratios being infinitely variable, this is a much larger range than is provided by any normal gearbox, and the higher ratios give the same sensation as free-wheeling.

Before describing the means by which the rollers are rocked, it is desirable to grasp the application of the foregoing principles in the actual gear, as shown in Fig. 197. The drive is transmitted by the clutch shaft (1) through dogs to the driving shaft (2). Floating on this shaft are the two outer races (3 and 4), being connected only by dogs for the drive. The steel rollers (5) are supported to the fixed roller assembly (6), which bolts between the two outer casings. The assembly consists of two sets of three rollers which can rotate on their own axes and, being mounted in carriers (15), can rock to various ratio positions. These transmit the drive from the two steel outer races (3 and 4) to the steel double inner race (7), from which it is conveyed by the large drum (8) to the propeller shaft (or, for reverse, through suitable gears).

As the transmission through the rollers and races reverses the direction of motion, the final drive in the rear axle has to be reversed by placing the crown-wheel on the off-side, instead of the near-side, of the bevel pinion.

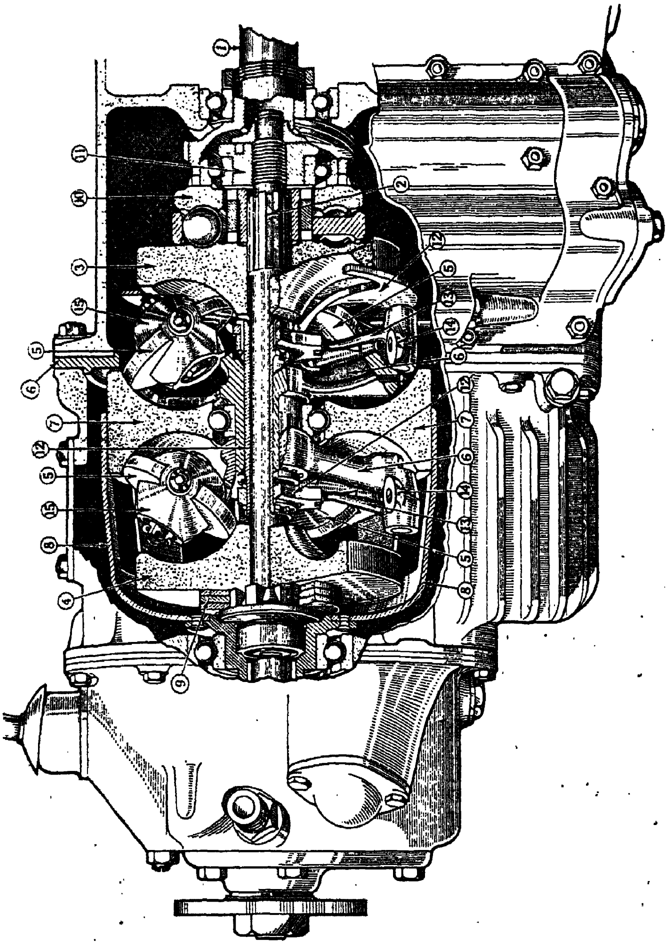
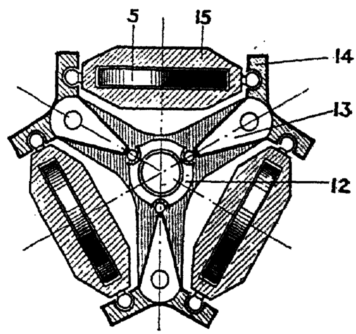


Fig. 197.—The Hayes Automatic Transmission.—The drive is transmitted by the outer races 3 and 4, which float on the drive shaft 2, through the rollers 5 to the centre race 7, from which it is taken by the drum 8 to the propeller shaft. The spring washers that provide the initial pressure on the assembly are shown at 9.

To transmit any given power a certain pressure must exist between the two sets of rollers and their four races. The initial pressure is obtained by three spring washers (9) giving approximately 1,400 lb. load on the roller. Further pressure is provided by a loading device incorporated in

the drive. The clutch shaft is dogged to a floating torque ring (10) and the outer race (3), as previously mentioned, also floats, but engages by means of dogs with the drive shaft (2). Grooves in the opposing faces of the torque ring and the outer race form inclines for three steel balls carried between the two members. These balls provide the drive by reason of their wedging action between the inclines. This action further serves to apply an end thrust on the entire roller mechanism, towards the rear outer race (4), which is supported by the spring washers (9) already mentioned. As the driving torque increases, the balls increase their thrust so that the rollers, being under a higher pressure, can transmit the heavier load. Incidentally,

FIG. 198.—A diagrammatic End View of the Roller Mounting showing the Means of Control. 12 is the control sleeve, 13 the rocker lever, 14 the rocker arm, and 15 is the carrier for the roller (5).



the thrust of the outer race being resisted by the spring washers (9) and the thrust of the torque ring by the loading nut (11), the entire thrust is sustained within the driving shaft (2) without imposing any unbalanced end load on any other part of the mechanism. Further, as there are three rollers between each pair of races the consequent three-point contact ensures uniform pressure on each. The pressure exerted on the roller renders it impracticable to change the ratio of the drive by directly rocking them, in view of the large effort required, the difficulty of moving all six rollers positively to exactly the same angle, and the slipping which would result from a sudden speed change. One of the most ingenious features of the Hayes Transmission is the way in which this difficulty is overcome. By altering slightly the axial position of each roller, it rolls of its own accord into a new path of contact. The sectional view of the transmission shows

the control sleeve (12) which by suitable levers is linked to the hydraulic control unit. This sleeve rotates to move the lever (13) shown more clearly in diagram form in Fig. 198. In this diagram, 12 is the control-sleeve round the main shaft, which operates the three rockers (14). These support the carriers (15) in which the roller (5) are mounted. It will

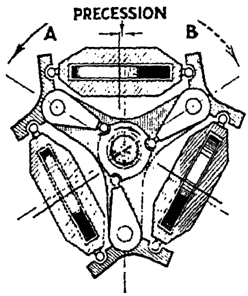


FIG. 199.—The Rear Roller Assembly as seen from the Rear, *A* is the direction of reaction due to tractive resistance. *B* is the reaction on over drive. The rollers are precessed for a low ratio.

be noted that the roller carriers have ball ends to provide a universal mounting in the rocker arms. The roller can therefore rock to any ratio position. Rotation of the sleeve (12) transmitted to the main rocker levers (13) causes the rollers to assume (for instance) the position shown in Fig. 199, and this displacement of the roller axes initiates precession of the rollers, which sends them to their new ratio position over a spiral (instead of their normal circumferential) path on the races. Thus, without involving any appreciable effort the rollers can be readily induced to take any ratio position required.

The hydraulic control unit comprises a pump, driven at half engine speed, to create oil pressure in the control cylinder on top of the control piston. As the oil pressure increases with the engine speed, movement is imparted to the piston which is communicated to the control sleeve to initiate precession of the rollers so as to give a higher ratio.

The driver's controls consist of two small levers above the steering wheel and a transmission lever to engage the forward or reverse drive.

The De Lavaud Variable Gear.—This gear, which has been fitted to Voisin cars, employs the principle of a swashplate of variable tilt (to obtain the variable gear

ratios), in place of the usual final bevel drive, and in its position. A number of connecting-rods are attached at equi-distant intervals to the periphery of the swash-plate, and at their other end operate ratchet gears, i.e., one-way clutches or free-wheels, on the back axle shafts.

In the earlier models four connecting rods were

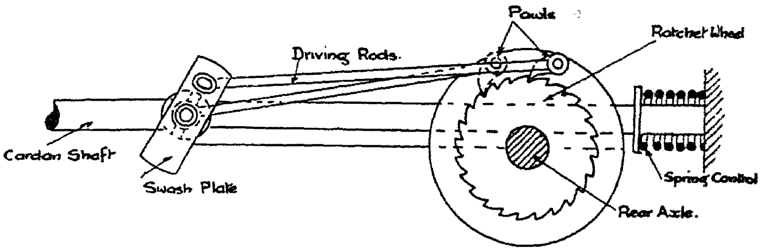


FIG. 200.—The Lavaud Variable Transmission Principle.

employed, but on later models six were fitted. The swash-plate is driven through an universal joint on the propeller shaft rear end, so that although it wobbles to and fro, when the propeller shaft rotates, it does not actually rotate. The connecting rods connected to it, are thus reciprocated to and fro, and their forward movement, in virtue of the ratchet gear, gives a positive turning movement to the back axle; their rearward movement is an idle, or free one, and does not rotate the back axle. The ratchet or free-wheel drive in the actual gear is made up of a series of rollers and cams or inclined plane devices somewhat on the lines of a bicycle free-wheel arrangement.

It will be evident that the greater the inclination of the swash-plate the greater will be the movements of connecting rods, and, therefore, the higher the back axle speed; this corresponds to raising the gear ratio. Smaller inclinations of the swash-plate correspond to lower gear ratios. In the later models there is an automatic device operated by means of a special spring at the rear of the back axle, for varying the tilt of the swash-plate, and, therefore, the gear ratio to suit the road resistance; this spring is deflected by a high road resistance—such, for example, as an upward gradient—and lowers the gear ratio.

Reverse is obtained through a special epicyclic gear. A differential action is obtained in virtue of the free-wheels in the rear axle.

Spicer design of universal joint shown in Fig. 205 was designed. The coupling member bearings are of the

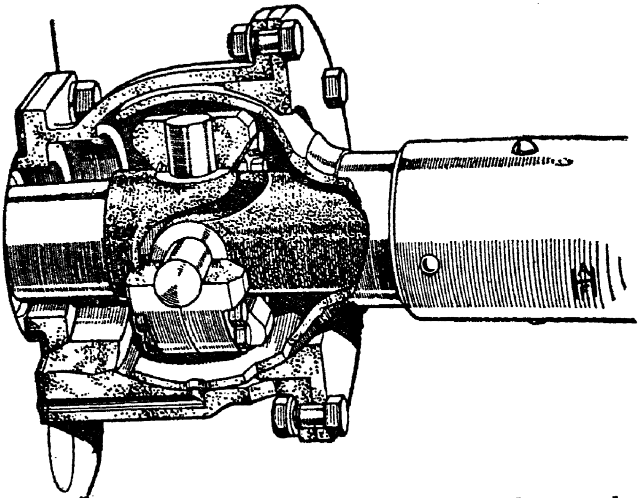


FIG. 204.—Type of Universal Joint that has been used on Morris Cars, with Enclosed Propeller Shaft Drive.

needle-bearing pattern and are entirely enclosed in a grease-packed mounting. The needle bearing consists of a relatively large number of small diameter hardened

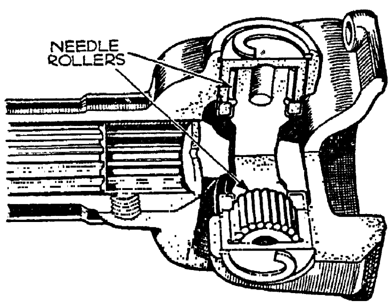


FIG. 205.—The Hardy Spicer Needle Bearing Universal Joint.

steel rollers which operate between hardened steel face on the joint pins and in the pin housings so that there is practically no wear experienced over long periods of usage

In most cases a greaser is provided for the splined shaft which slides in the rearmost universal joint member's corresponding splines.

In another design of universal joint made by the same firm, the needle bearings are replaced by special rubber bushes, in which the tapered coupling pins are inserted; the relative movements between the pins and their bearings are then taken up by the twisting action of the rubber bushes as in the Silentbloc rubber bearings.

An Improved Universal Joint.—The ordinary Hooke's coupling when applied to connect a pair of inclined shafts, one of which is driven at a uniform speed of rotation—as with the gearbox shaft—does not produce a uniform

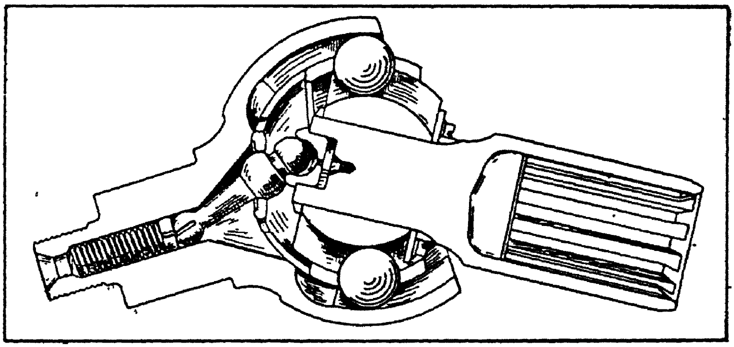


FIG. 206.—A Constant Velocity Universal Joint.

angular velocity for the driven shaft; the latter varies in velocity throughout its revolution. If, however, a pair of parallel shafts are connected by an inclined shaft having Hooke's couplings at each end and one shaft be driven at a uniform speed, then the other parallel shaft will also run at the same uniform speed. In order to overcome the disadvantage of uneven speed in the case of a single Hooke's coupling, an improved joint, known as the 'Rzeppa' has been devised. In this case, a compensating device (Fig. 206), is incorporated to give uniform velocity to the driven inclined member over the usual range of propeller shaft working angles. This coupling uses large diameter balls for transmitting the drive between the two shafts.

Fabric Universal Joints.—Another type of universal joint, or coupling between the gearbox and propeller shafts that was once more widely employed is that known as the *Flexible Fabric Disc* type.

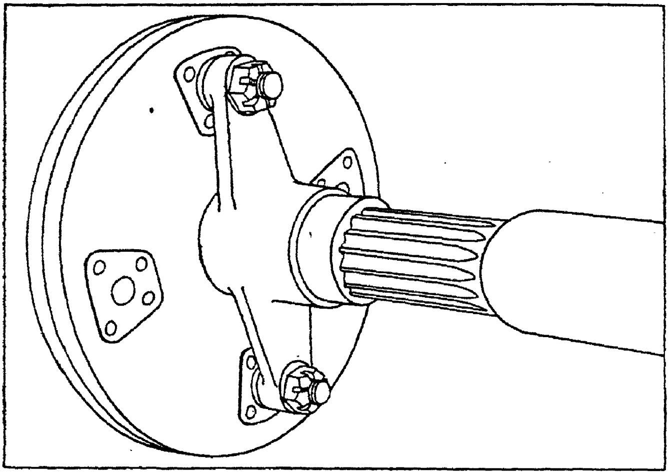


FIG. 207.—Fabric Universal Joint with Sliding Splined Shaft.

In this case both the clutch and propeller shaft ends are provided with arms or 'spiders,' as they are termed. Each spider consists of a pair of arms placed oppositely on the axis of its shaft. The two spiders are arranged, in their working positions, so to be at right angles to one another, as in the case of the Hooke's coupling. Instead of a bearing ring, two or more annular discs of rubberised canvas, of much strength and flexibility are employed to connect the spiders, by means of stout bolts and metal plates or washers, as shown in Fig. 207. The propeller shaft can work at an angle to the gear shaft, in virtue of the elasticity or flexure of the fabric joint. In the usual design the spiders have two opposite arms, but in other cases, three equally spaced arms are provided on each spider.

For a 12 h.p. (R.A.C. rating) car, flexible coupling, there would be two $\frac{3}{8}$ in. thick (or three $\frac{1}{4}$ in. thick) annular discs of 6 inches outside diameter, with a 2 inch hole. The discs would be secured to the spiders by $\frac{3}{8}$ or $\frac{7}{16}$ inch diameter bolts.

The advantages of the flexible fabric universal, are its complete silence in action, long life, owing to absence of wear, and non-lubrication attention.

Rubber Trunnion Block Universal Joint.—Another type of universal joint which has more recently become popular is the Layrub one (Fig. 208). This consists of a two-hole flange coupling on each of the driving and driven members, and an intermediate metal disc member having four holes locating large flexible rubber blocks. The two coupling flanges of the joint are at right angles and bolts connect

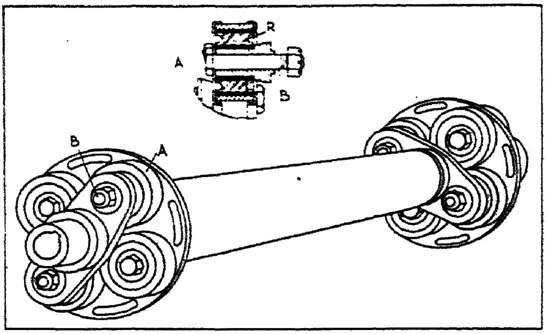


FIG. 208.—The Layrub Universal Joint.
(A) Trunnion Block. (B) Bolt.
(R) Rubber.

the flanged parts to the rubber blocks. The inclination of the propeller shaft is taken up by the yield of the rubber trunnion blocks; the latter have embedded fabric on the insides of the holes and on the peripheries. It is claimed that, owing to the axial yield possible with a pair of such couplings it is unnecessary to use a sliding splined shaft. There are no metal-contacts in this coupling, the drive and inclination being taken by the rubber trunnions, so that torsional vibrations are absorbed and a quiet drive results. No maintenance is required with this type of coupling as there are no rubbing parts requiring adjustment or lubrication.

The Sliding Coupling.—When the rear axle unit moves up and down under the springing action, since it is connected to the chassis at the front end spring bearings, it will move

in an arc of radius approximately equal to the distance between the centre of the rear axle and the spring pin centre. Since, however, the propeller shaft will tend to swing up and down about a centre near the rear end

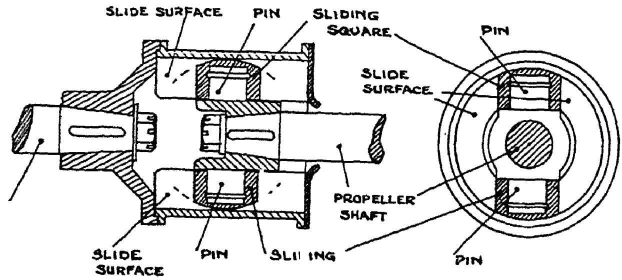


FIG. 209.—Plunging, or Sliding Type Universal Coupling.

of the gearbox, i.e., at the front universal joint, it will have to lengthen and shorten as the rear axle swings about the spring centres. In order to allow for this tendency a sliding joint is invariably provided in open-propeller shaft systems of the type under consideration.

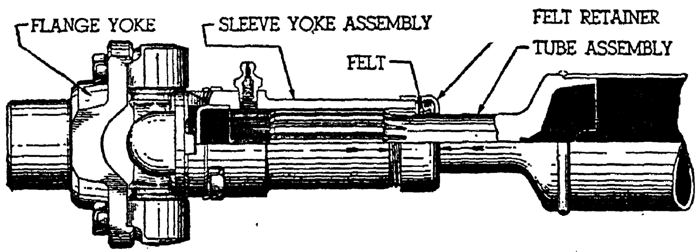


FIG. 210.—Sliding Splined Tubular Propeller Shaft with Universal Joint Member.

In the earlier model cars the rear end coupling was combined with the sliding member as shown in Fig. 209; it was known as the 'plunging' or 'pot' coupling. In this example a forked arm fitted to the rear end of the propeller shaft, was provided with pins, one on each side. On these pins square shaped steel members could rock. The fork arm with its square members could slide in a pair of slots, oppositely placed, in a

cylindrical member secured to the rear axle drive. The outside surfaces of the square members were spherical in shape, so that the propeller shaft could rock as well as slide whilst transmitting the power. Unless well lubricated, and enclosed in dust-tight leather or fabric covers, this type of joint was apt to wear, and to become noisy in action.

The modern method of allowing for the sliding movement of the propeller shaft is to spline the front end of this shaft and allow it to slide in corresponding splines cut inside the front universal joint member, as shown in Figs. 207 and 210, as previously mentioned a greaser is provided to lubricate the splines, grease-gun injection being employed for this purpose.

The open propeller shaft drive with its universal and sliding couplings is sometimes referred to as a *Live Axle Transmission.* The maximum angle of movement of the propeller shaft in either case is usually from 12° to 18°, whilst the average sliding movement is about ½ to ¾ inch on bad roads.

Enclosed Propeller Shaft.—Another propeller shaft arrangement in present use, consists in enclosing the

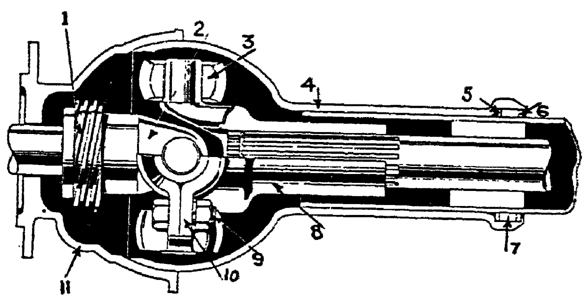


FIG. 211.—A Typical Front End Universal Joint for Propeller Shaft.

1. Speedometer drive gear.
2. Universal joint yoke.
3. Universal joint block.
4. Torque ball.
5 and 6. Torque ball gasket retainers.
7. Torque ball gasket.
8. Universal joint driven yoke.
9. Universal joint ring bolt.
10. Universal joint ring.
11. Gearbox end plate.

shaft in a tapered tubular casing, connected rigidly to the back axle casing. The front end of this propeller

shaft casing is provided, either with a forked or a spherical joint, which works in pin bearings, or in a spherical bearing mounted on the frame cross-member just behind the gear-box (Figs. 204 and 211) respectively. In this case, the propeller shaft moves bodily with its casing under the action of the rear wheel spring movements, it is therefore, unnecessary to provide a sliding coupling at its rear end, but only an universal coupling at the front end, working inside the ball-joint of the casing ; the thrust of the rear wheel drive is then transmitted to the chassis frame through this casing and ball-joint, leaving the springs to look after the springing action only.

A typical forward-end metal universal joint enclosed inside a ball-ended propeller-shaft casing is shown in Fig. 211. The flange on the left-hand side of the casing 11 is for bolting this member to the rear end of the gearbox.

The universal joint itself is shown at 2 and 3, whilst

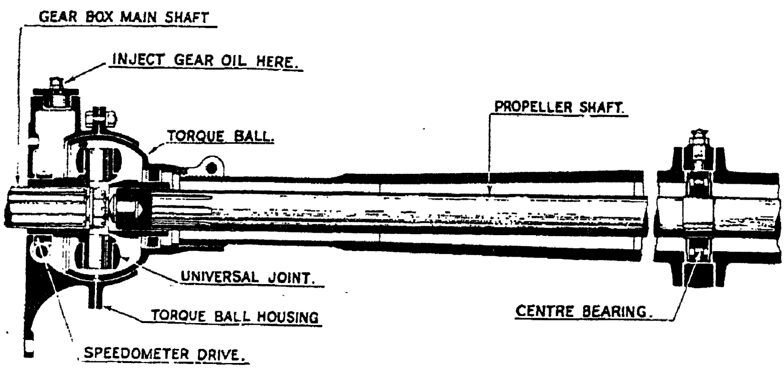


FIG. 212.—Showing Universal Joint and Intermediate Bearing for Propeller Shaft.

the spherical torque tube bearing is at 4. This joint, and also the universal joint is lubricated automatically from the gearbox. The ball-and-socket type bearing, in this case, allows the torque tube and propeller-shaft to move up and down as the rear axle moves under the influence of the rear springs. The shaft itself has a 'steady' bearing at 5 and 6.

Long Propeller Shafts.—In the case of long propeller shafts it becomes necessary to prevent the flexing or 'whip'

of the shaft, otherwise excessive wear may occur. One method that is used on o vehicles having enclosed propeller-shafts is to fit a centre ball or roller bearing in the torque tube, as shown in Fig. 212. This bearing is provided with a gre lubricator.

Another method is to *divide the p* into two parts, the front portion being between : and one of the cross-members of the frame, in a suitable ball or roller bearing is fitted. The rear portion is often made the same as the usual torque tube, with ball end bearing and enclosed propeller-shaft.

It is advisable to fit a universal joint to the front end of the front shaft, i.e., at the gearbox, and the usual pair of universal joints to the rear shaft.

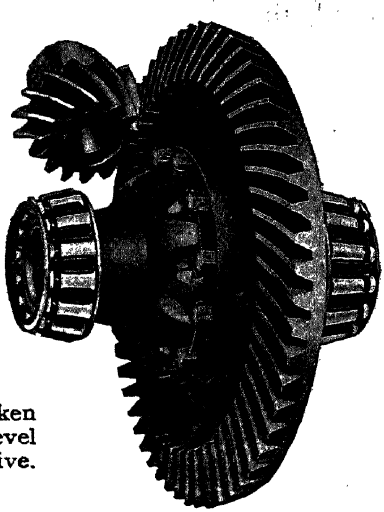


FIG. 213.—The Timken Spiral Bevel Final Drive.

The Final Drive.—The engine crankshaft being arranged at right angles to the rear axle, it follows that at the back axle some form of gearing must be provided to transmit the power to each of the rear wheels, and at right angles to the propeller shaft. There are several forms of gearing which allow power to be transmitted from one shaft to another at right angles to it, and without any appreciable loss of power. The final drive gear is designed to give a reduction ratio of between 4 : 1 and 5 : 1 in modern cars, so that the rear wheels are thus driven at $\frac{1}{4}$ to $\frac{1}{5}$ of engine speed when top or direct

drive gear is engaged in the gear box. The principal types employed in motor-car practice are (1) The *Bevel Gear*, and (2) The *Worm and Worm Wheel*. In the case of the bevel gear, which consists merely of two toothed cones (or frustra of cones) meshing together, their axes meeting in a point, a smaller bevel gear (Fig. 213) attached to the rear end of the propeller shaft or its flexible coupling, meshes with a larger bevel wheel, known as the *Crown Wheel* which is attached to a gearing driving the two rear wheels. In former days bevel gear had straight teeth (*i.e.*, teeth which if produced would meet at the common point of intersection of the two rolling toothed cones). Unless, however, the teeth were cut very carefully, and the proper depth of engagement of the teeth was arranged, there was not only much noise when the

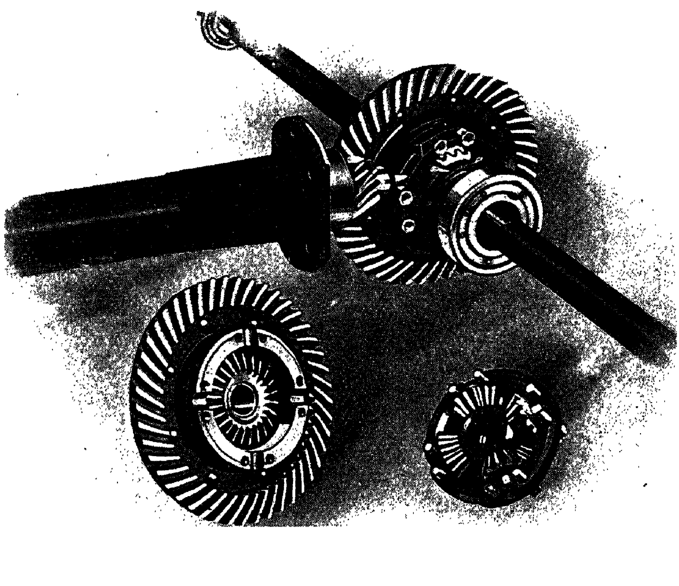


FIG. 214.- -Helical Bevel Car Final Drive, showing Differential Gear and Back Axles.

gears were working, but a loss of power. More recently the *Helical Bevel* gearing has superseded the straight tooth type. In the latter gear any tooth goes into mesh at one time all along its length, whereas in the helical

bevel form of gear, the meshing commences at one end of a tooth and travels towards the other end. The teeth in this case are cut on the slant, or cross-wise as shown in Fig. 213. It will be seen that, apart from the gradual meshing of a single tooth, two or three helical teeth are in partial mesh at the same time. These gears, therefore, work very silently, efficiently and without backlash, even after long periods of running.

There is another type of bevel gearing, in which the teeth are arranged in *chevron* fashion, each tooth being composed of two portions inclined one to the other; in effect the *Chevron or Herring Bone Gear* consists of two helical bevel gears inclined to each other. They are very quiet in action and efficient. The gear which resembles that shown in Fig. 163 is known as the Citroen type; it was employed for the final drive on Citroen cars.

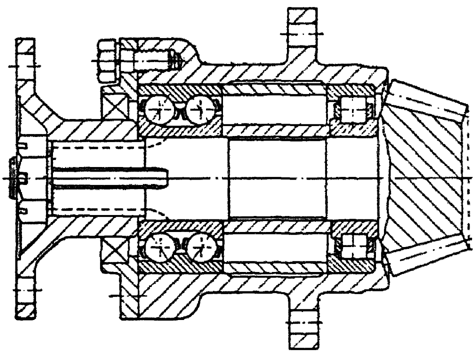


FIG. 215.—Bevel Pinion Gear Shaft Bearings.

The Bevel Pinion Bearings.—It is most important that the bevel pinion should be maintained accurately in position relatively to the crown wheel so that the gear teeth are meshed correctly under all running conditions. Since the bevel pinion unit is subjected to end thrust loading it is necessary to provide some form of thrust bearing. It is usual to employ either double row deep ball bearings capable of taking end thrust in either direction or a pair of opposite inclined roller bearings, as shown in Fig. 217. A recommended arrangement is to employ a roller bearing close to the bevel gear and a double-race thrust and journal Skefko ball bearing at the other end (Fig. 215).

With this arrangement neither end play nor deflection can occur, the gears meshing accurately all the time.

Where exceptionally heavy torques have to be trans-

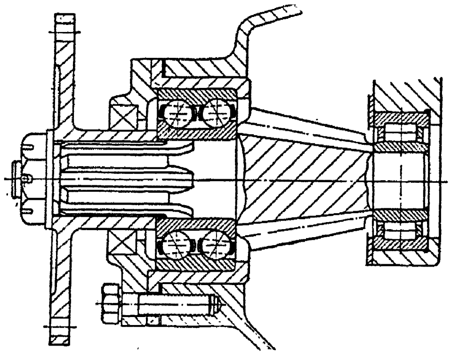


FIG. 216.—Straddle Mounting for the Bevel Pinion Gear Shaft.

mitted, as in high-powered cars and commercial vehicles it is usual to provide an outside bearing for the bevel pinion shaft. This is known as a *Straddle Mounting*, a typical example of such a mounting being shown in Fig. 216.

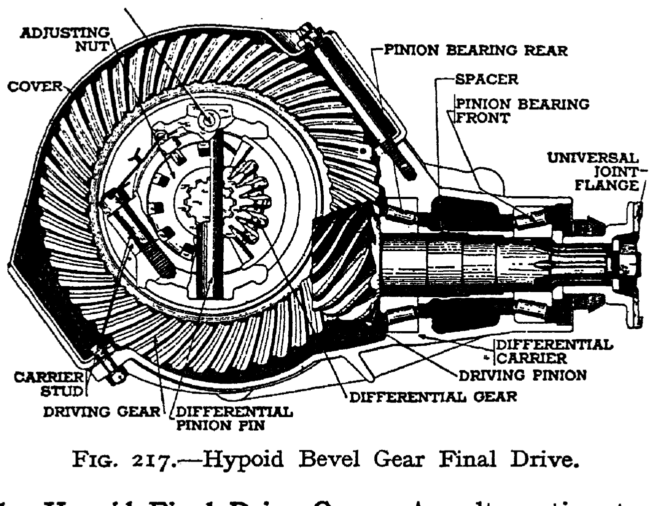


FIG. 217.—Hypoid Bevel Gear Final Drive.

The Hypoid Final Drive Gear.—An alternative to the symmetrical bevel gear final drive is that known as the

Hypoid Gear, for which the advantages of quieter running and a lower propeller shaft position are claimed. Fig. 217 shows a typical hypoid gear as fitted to certain cars of American origin. It will be observed that the final drive pinion, which is mounted in opposed inclined roller bearings has its axis of rotation about halfway below the centre of the crown-wheel and its periphery.

Worm Drives.—Another exceedingly quiet and very efficient form of final drive is that in which the propeller-shaft drives a steeply pitched screw, or *Worm,* which meshes into a *Worm Wheel,* having its axis of rotation coincident with those of the rear wheel axles. Very careful design and manufacture is necessary, but the result is probably the most efficient and the quietest type of gearing known. Some tests on Lanchester car worm gears showed that only 2 to 4 per cent. of the power transmitted was lost. In the case of well-designed bevel gears, the average loss is from 3 to 5 per cent.

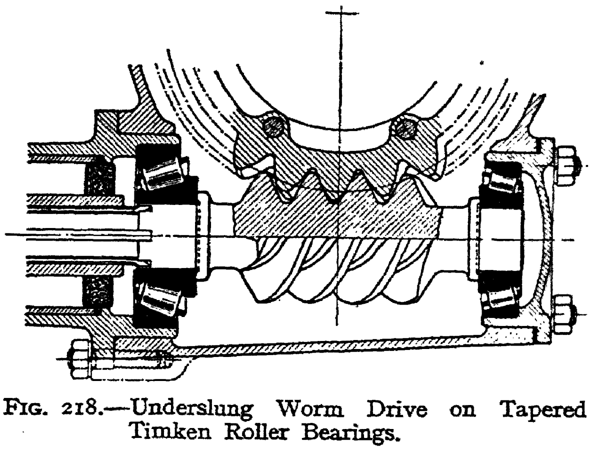


Fig. 218.—Underslung Worm Drive on Tapered Timken Roller Bearings.

The more recent Holroyd-Walker worm gear, embodying a special thread shape gave an efficiency of 97·6 per cent. under various conditions of speed and power, when tested at the National Physical Laboratory. This was the highest recorded efficiency for worm gears.

There are two principal types of worm, viz., the *straight* or Brown type, and the *hour-glass* or Lanchester form. In the former case the worm is cut from a cylinder and can be

adjusted endwise without altering the meshing of the gears. In the latter case the worm is of hour-glass form, and is made to follow the contour of the worm-wheel. It requires rigid supports at its ends and must be accurately positioned. The worm usually has two or three separate 'starts' or threads and the *pitch* is chosen so that one revolution of the worm rotates the wheel one-quarter to one-fifth of a revolution. The worm is usually made of hardened steel, and the worm-wheel is of phosphor-bronze or gun-metal.

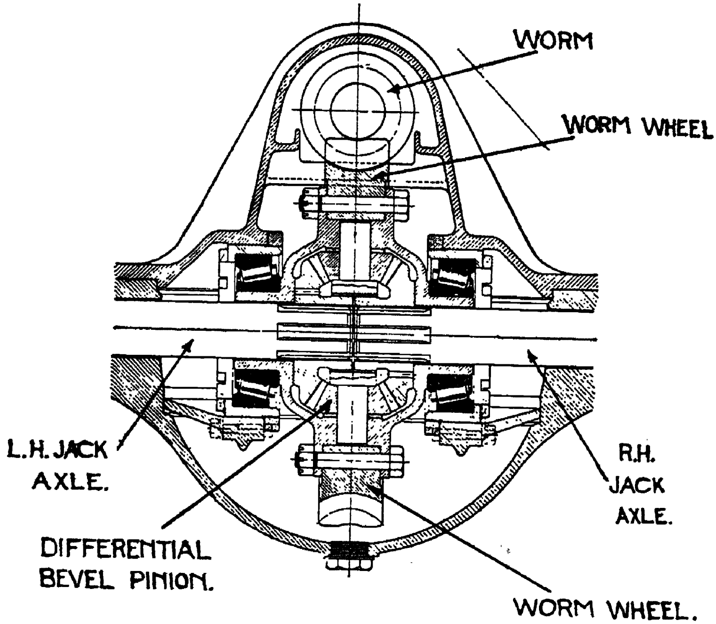


FIG. 219.—Overhead Worm and Wheel Final Drive and Differential Gear.

When the worm is arranged *on top* of the worm-wheel it is known as an *Overhead Worm Drive*. It has the advantage of giving greater ground clearance to the back axle casing, but it is more difficult to lubricate; cases have occurred in certain small car models of excessive wear in worm drives through lack of lubricant.

Fig. 219 shows an overhead worm final drive design in which the crown or worm wheel is mounted on Timken tapered roller bearings. The outer races of these bearings are housed in the back axle casing and the inner races

are a tight fit on the worm member casing which carries the differential gear pinions shown.

When the worm is underneath the worm-wheel the arrangement is known as the *Underslung Worm Drive.* Although giving reduced ground clearance, it ensures adequate lubrication of the worm and bearings.

Fig. 218 shows an underslung worm arrangement for a light car, using Timken type roller bearings; whilst Fig. 219 shows in front and side views an alternative arrangement for a medium car. The differential pinions and worm-wheel are clearly shown in these illustrations.

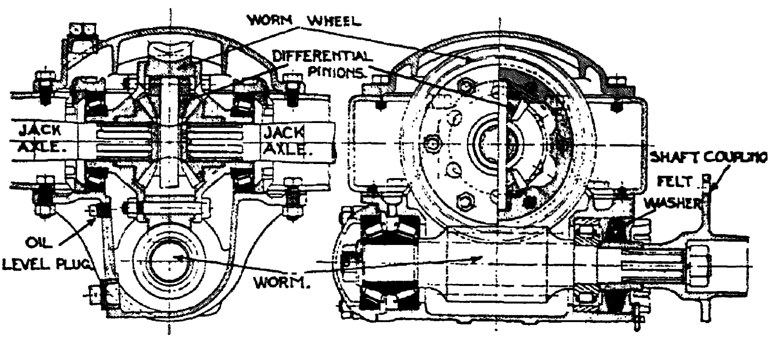


FIG. 220.—Front and Side Sectional Views of Underslung Worm Driven Axle.

End Thrust.—In the case of bevel gear final drive, there is an endwise thrust exerted on both bevels, tending to force them axially. It is, therefore, necessary to provide thrust bearings on the shafts of both bevels.

When the final drive is by worm and worm-wheel, there will be a rearward axial end thrust on the worm when the car is moving forward, in the case of a right handed overhead worm, and a forward thrust when in reverse. Ball or roller end thrust bearings to take the thrust both ways must, therefore, be provided for the worm-wheel.

The Differential.—The power transmitted by the final drive must be divided and delivered to each of the rear wheels. In the simplest possible case, the crown-wheel of the bevel drive is keyed to a single continuous axle, which can rotate in four ball bearings, two at

the central portion, and one at either end of the back axle casing. The road wheels in this case are forced on to the tapered ends of the axle shaft by means of a screw thread and nut, keys being provided to prevent slipping whilst driving. In this case both of the road wheels must rotate at the same speed.

The important feature of the construction described is the solid back axle, to which the rear wheels are keyed, so that they both turn at the same speed. Now we have seen in Chapter III that when the car is steered to the right or the left, all of the wheels are momentarily turning about a common (or instantaneous) centre, and, therefore, at different radii. The outer rear wheel on the curve must consequently tend to rotate faster than the inner rear wheel, owing to its greater turning

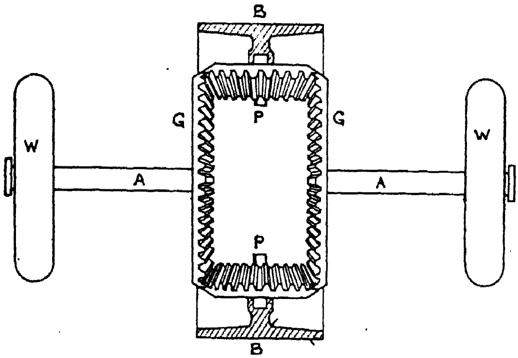


FIG. 221.—Simple Diagram to Illustrate the Principle of the Differential.

radius: if the back axle and wheels are integral as in the preceding example, the outer wheel will tend to skid when turning corners, or both wheels may skid a little due to the different speeds. On a 35 foot radius turning circle, which is a common value for medium cars, the outer wheel will tend to turn about 13 per cent. faster than the inner wheel.

By keeping the wheel track small, and the turning circle radius large, the skidding effect can be reduced to small proportions. Experience with solid rear axles on earlier small cars does not appear to indicate any appreciable extra tyre wear, due to the solid axle; this

may be due to the small percentage of the total mileage which is accounted for by appreciable turns.

In order that each of the two rear wheels may rotate at its correct speeds when negotiating corners, it is necessary to introduce a special type of gearing, known as the *Differential* between the two separate halves of the back axle shaft; each of the latter is known as a *Jack* or *Half Shaft*. The object of the differential is to enable either jack shaft to rotate at a different speed to the other, when necessary, and whilst transmitting the drive. The

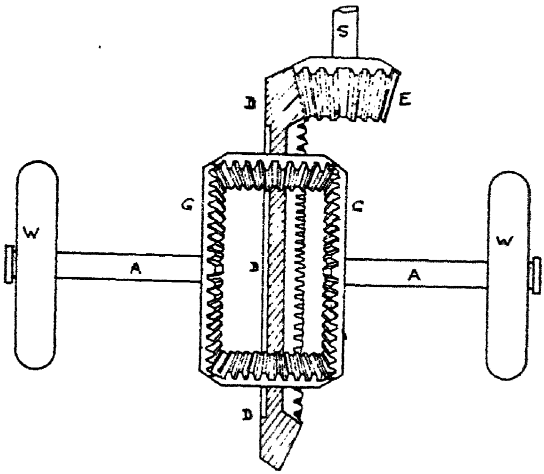


FIG. 222.—Operation of the Differential Gear.

principle of the differential is illustrated in Figs. 221 and 222. For the sake of simplicity, a smooth belt pulley *B* is shown in Fig. 221, corresponding to the crown wheel of the final drive. The pulley *B* carries the bearings of two bevel pinions *P*, the latter being free to rotate in these bearings. As the pulley *B* rotates it carries the bearings of the pinions *P* around with it, and, therefore, at the same speed. It will be evident also that if both of the wheels *W*, attached through the jack shafts *A*, to the bevel pinions *G*, are fixed, the pulley *B* cannot turn, but will exert an equal turning effort or force on each wheel *W*. If both wheels *W* are free to rotate, and each offer an equal resistance, then it is clear that the pulley *B* will drive the pinions *P* and bevel wheels *G* around solidly, as though they

were locked; the two wheels *W* will, therefore, rotate at the same speed, as though these were a solid back axle. Next, imagine that the left wheel *W* is fixed solidly; a little consideration will show that the pinions P will roll round the left bevel wheel *G*, and will cause the right wheel *G* to rotate, positively and at the speed of *B*. Thus, it is possible to drive one wheel *W* at normal speed whilst the other is fixed. It will now be evident that since we can drive both wheels at the same speed, or can fix one and still drive the other, without altering the form of drive, that if both road wheels are free, one having a greater resistance to its motion, the differential gear described will enable it to go slower than the other. This is exactly what happens during a turn, for the inner wheel tends to drag, and slows down, whilst the outer wheel keeps its speed. Therefore, we have, in the differential, a positive drive which enables either road wheel to rotate at a different speed to the other.

Referring to Fig. 222 it will be seen that the same explanation will apply when the pulley *B* is replaced by the crown wheel *D* and final drive pinion *E*.

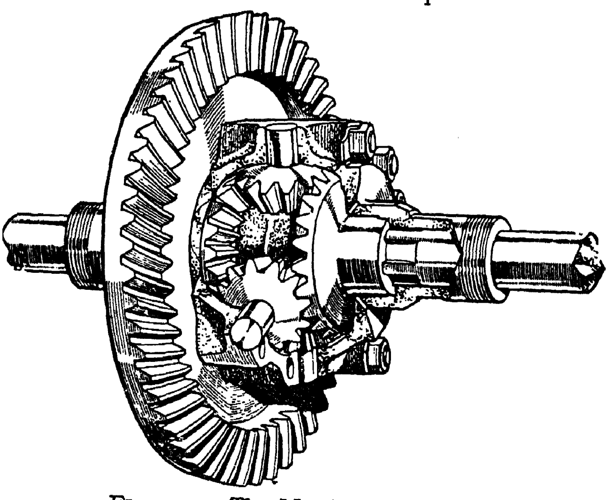


FIG. 223.—The Morris Differential Unit.

Typical Differential Gears.—There are several types of differential gears in use. The majority, however, empl bevel pinions for the differential gears, with bevel

and crown-wheels, or worm and worm-wheels for the final drive.

Fig. 223 shows the differential assembly of the Morris cars. The spiral bevel crown-wheel carries a casing in which the differential bevel pinions are housed. One of the driving bevel pinions, viz., that on the right-hand jack axle-shaft, is also shown. The driving bevel pinion for the crown-wheel has been omitted for the sake of clearness.

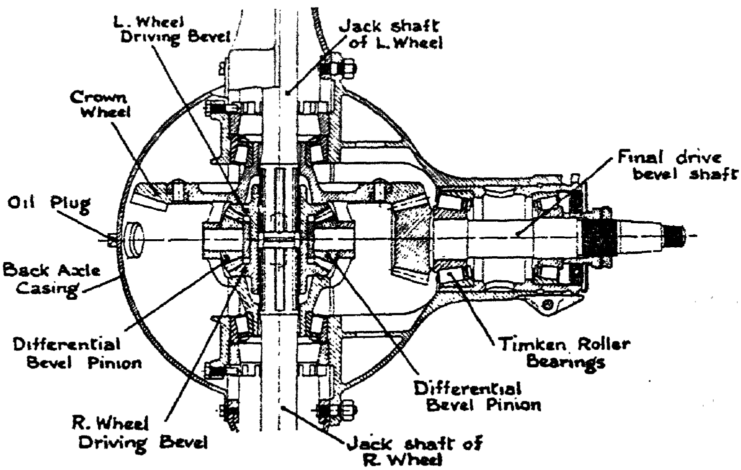


FIG. 224.—A Common Differential Gear Arrangement.

Fig. 224 shows a complete differential unit of the bevel pinion final drive type.

Instead of bevel pinions, ordinary straight pinion wheels have been used, in the past, for differential gears; the Wolseley car formerly employed this type.

Testing the Transmission.—It is as well to note that if *one of the rear wheels* of a motor-car *be jacked up*, the other remaining on the ground, and the engine be running, by engaging one of the gears in the gearbox, and letting in the clutch, *the wheel which is jacked up will rotate*, the other remaining stationary; in this way tests of the transmission may be made.

Fig. 224 shows the layout of a typical back axle and differential casing. It will be observed that tapered roller bearings (Timken) are fitted to the final drive bevel shaft, and also to the crown wheel, and differential bevel

casing. These bearings take both the ordinary journal loads and also the driving end thrust of the bevels. The two jack shafts are provided with splined ends, which fit into the differential driven pinions (corresponding to

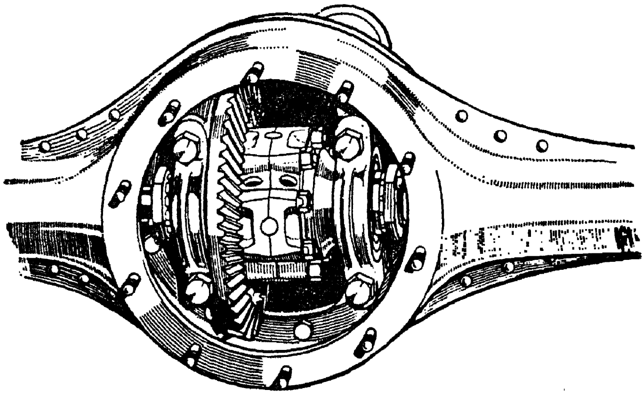


FIG. 225.—Back Axle Casing, with Cover removed to show Differential Gears.

G in Fig. 222). The hardened alloy-steel crown wheel is riveted to the casing carrying the differential pinion gears; the latter is usually a gunmetal casting. The differential gear is sometimes provided with straight toothed, or spur wheels, instead of bevel pinions, the wheel driving gears being also of the same type, as previously mentioned.

Some Peculiarities of the Differential.—We have seen that the drive will be transmitted entirely to one wheel if the other is fixed; this result occurs when one wheel is jacked up, or when one rear wheel brake acts before the other. Again, if one of the rear wheels happens to sink into soft mud, whilst the other is on firm ground, there will be little resistance offered by the former, so that it will spin round, whilst the other remains stationary. It is therefore impossible to drive the car out of such a position, except by attaching a chain around the wheel which is in the mud, so as to increase its resistance. One or two designs provide means for locking the differential gear in such cases so as to obtain a solid back axle.

Another peculiar effect, observed in the case of certain taxi-cabs, was the locking of one wheel when the brake was applied, and the reverse rotation of the other. Usually

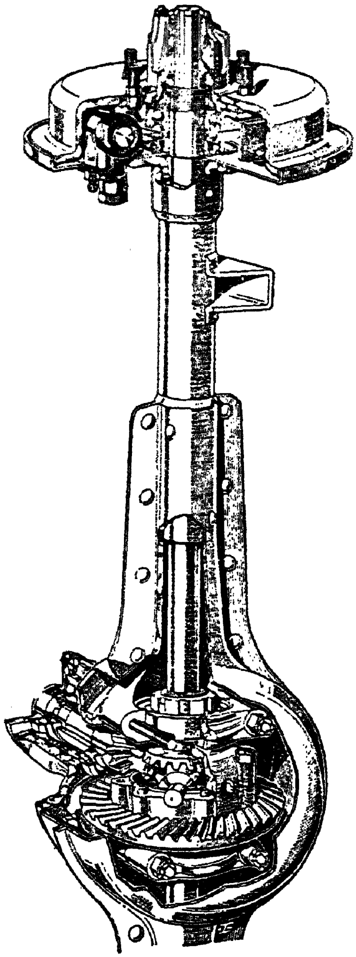


Fig. 226.—Morris Back Axle Arrangement.

this effect was due to the fitting of a brake on the propeller shaft, just to the rear of the gearbox. When the brake was applied, it locked the final drive bevel, the crown wheel and the differential pinion casing. The differential pinions could still rotate in their bearings, however. If,

as is often the case, one of the rear wheels tends to slip on the road more than the other due to its lower resistance to turning, then the one which does grip the road will cause the differential bevels P (Fig. 221) to turn, and these will cause the other wheel of lower resistance to *rotate in the opposite direction* to the other wheel; this effect often occurs when one rear wheel has a steel studded tyre and the other a plain tyre; the former is the one which reverses its rotation, on hard macadam or asphalt roads.

The Back Axle.—The purpose of the unit known as the *Back Axle*, is to enclose the final drive, whether bevel or worm and worm-wheel, the differential gear, and the axle shafts, together with their bearings in a dust and oil-tight casing. The back axle serves also to locate, or position the bearings for the gearing and axle shafts; it carries also the spring pads, by which the rear springs are attached, the radius rods (if fitted), the propeller shaft casing, and the rear wheel brakes.

Fig. 226 illustrates a typical, well-designed back axle, and its components. The main casing is of pressed steel, reinforced for strength; this gives a very strong yet light, form of construction. In some cases the main portion of the back axle casing is a solid forging, machined

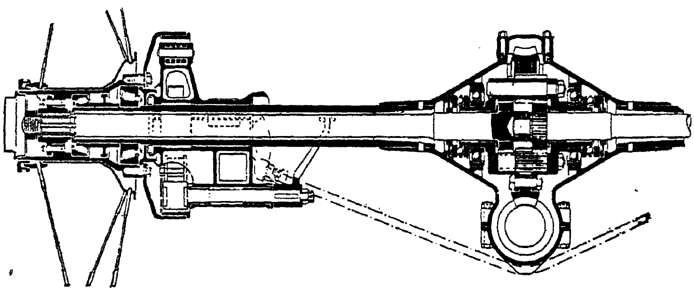


FIG. 227.—Another Back Axle Design, with Underslung Worm Drive, and Straight-toothed Bevel Pinions.

out; in others it is built up of malleable iron, or aluminium alloy castings. The front cover is a casting in aluminium alloy, or iron, and carries the final drive shaft bearings. The rear cover is a dome-shaped circular steel pressing, bolted to the main casing, as shown in Fig. 224.

By removing this cover, the differential gears may be inspected or removed without dismantling the rear wheels.

In one or two cases, the rear axle is constructed of thin steel pressings, built up on to the central and end members.

Fig. 227 illustrates another type of rear axle construction. This final drive is by an underslung worm and worm-wheel. Straight-toothed spur pinions are employed for the differential; two of these are shown.

The large journal type ball-bearings for the differential casing, and the thrust washers on either side will be noted. The back axle casing consists of a central iron casting, housing the other bearings mentioned, and also those of the worm-drive. Steel tubes are forced into this casing

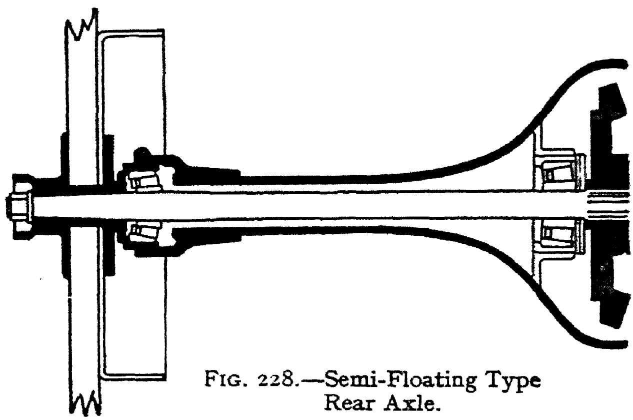


FIG. 228.—Semi-Floating Type Rear Axle.

and riveted in position, to connect it to the wheel-carrying portions; the latter include both the brake casings and the spring pads.

Types of Back Axle.—There are three principal types of back axle, namely, the *Semi-Floating*, *Three-Quarter Floating*, and the *Full-Floating* types. In the best practice the latter type is employed, although either of the other types can be designed to give equally good results.

Referring to the diagrammatic sketches in Figs. 228, 230 and 231, the former represents the *semi-floating* type. In this case the axle or jack shaft, is carried in roller bearings in the rear axle casing, the road wheel being attached

to the tapered end of the axle shaft by means of a nut and washer. The wheel is completely overhung, in respect to the bearing, and the whole of the weight of the rear portion of the car comes upon the axle shaft ; the latter are, therefore, subjected to bending as well as torsion stresses.

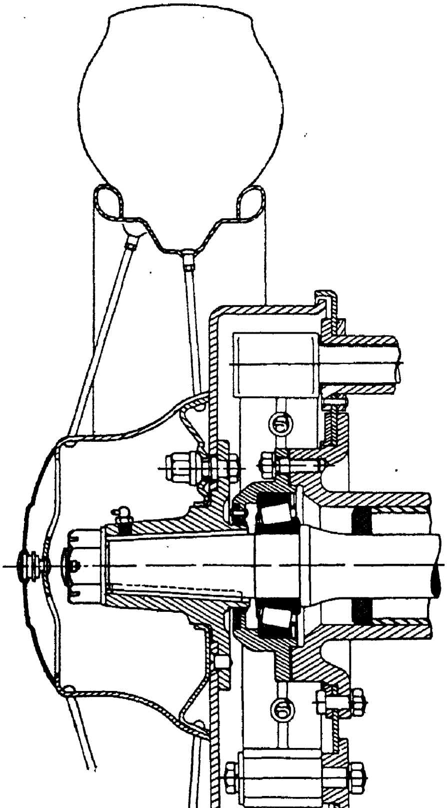


FIG. 229.—Semi-Floating Rear Axle, with Brake Drum and Wire Wheel.

Fig. 230 illustrates the *three-quarter floating* type, in which the axle shaft bearing is directly under the centre of the road wheel, and, therefore, takes a part of the weight. This type is sometimes termed the 'flanged-shaft' type, owing to the design of the wheel hub.

The *full-floating* type of axle is shown in Fig. 231. In this case the rear wheel is carried entirely on two sets of bearings housed on the back axle casing extension ; the

latter, therefore, takes the weight of the rear part of the car. The axle shaft now transmits the drive only, and so is not under bending action due to the weight. The outer end of the axle shaft is usually provided with

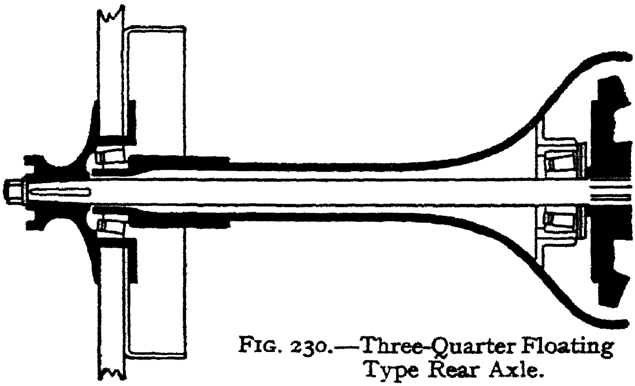


FIG. 230.—Three-Quarter Floating Type Rear Axle.

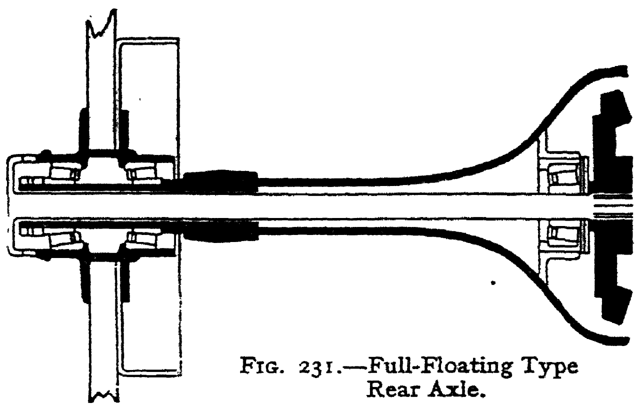


FIG. 231.—Full-Floating Type Rear Axle.

splines engaging with a small coupling; the latter has a number of teeth, or dogs, which engage with corresponding members machined in the wheel-boss flange. The axle-cap and usually a compression spring hold the coupling teeth into mesh with those of the wheel flange. In this design, *the axle shafts can be removed without jacking the back axle up*, or removing the differential gear; they can be pulled out, after the axle cap is removed. In some designs the flange, or coupling on the axle shaft, is bolted to the wheel-hub flange.

Rear Wheel Thrust Bearings.—It is necessary to provide thrust bearings for the rear wheels, for the latter are subjected to side forces when the car is moving over uneven ground or around curves. Taper roller bearings, or ball-thrust bearings are usually employed for this purpose.

Transverse Spring Back Axle.—In the case of back axles having the transverse method of springing, as in the Ford cars, it is, of course, necessary to position the back axle independently of the spring locations, in order to prevent the axle unit from twisting about a central vertical axis.

This positioning of the back axle is arranged in the case of the Ford back axle by means of the triangular bracing system shown in Fig. 232. The spring-end brackets on

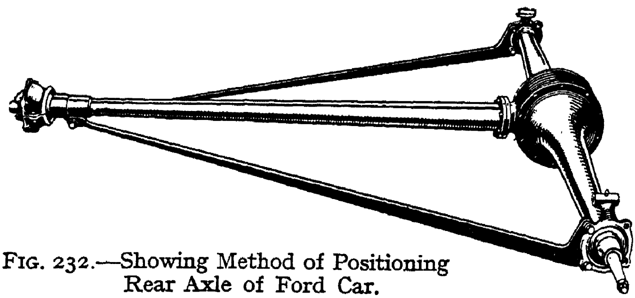


FIG. 232.—Showing Method of Positioning Rear Axle of Ford Car.

the back axle casing are connected by tie-rods—or *Radius Rods*—to the front end of the propeller-shaft casing, so as to form a rigid unit. The back axle cannot, therefore move relatively to the propeller-shaft casing.

Back Axle Tie Rods.—In certain cases of built up back axle casings, these were further stiffened against downwardly bending action by the provision of a bent tie-rod, resting at its middle against the lowest portion of the central casing, and attached at its ends by means of threaded portions and nuts to suitable lugs on the outer ends of the casing. In effect this gives the well known 'king-post and ties' method of stiffening. The dotted lines in Fig. 227 illustrate an example of this bracing method.

Care of the Back Axle.—As a rule little attention is necessary for the back axle. It is, of course, required to keep the casing supplied with the type of lubricant recommended by the makers. In most cases there is a lubricant level plug fitted to indicate the correct level of the lubricant. It is advisable to inspect the level, and to refill if necessary, once every 4,000 to 5,000 miles of running, according to the type. Any play in the hub bearings should be rectified, and in the case of semi-floating axles the alignment of the wheels on their axle shaft cones should be checked by jacking up each wheel in turn, and spinning. One occasionally observes cases of rear wheel wobble due to the wheel not fitting the tapered shaft properly, or to ts riding on the key.

Any excessive oil consumption in the back axle may be due either to a leaky central joint, to a loose base-plug, or to defective oil-retaining felt washers at the ends of the axle-shaft bearings. This oil leakage should be seen to without delay as the oil may find its way on to the brakes, causing slipping.

Noise in the differential pinions can, in some designs, be obviated by screwing their bearings inwards, so as to mesh the teeth a little more closely. In some cases a screwed split housing for the inner jack shaft bearings is fitted for this purpose, the split portions being locked by clamping together. A collar is provided on each of the screwed bearings; tommy-bar holes being drilled around the periphery for adjusting purposes.

Fuller information on this subject is given in Volume 4 of this series on "Car Maintenance and Repair."

CHAPTER VIII

THE BRAKES

Some Theoretical Considerations.—It is necessary to provide means for bringing the car rapidly to rest from any speed. Although in the ordinary way if the engine is switched off, and the gears are left in mesh in the gearbox, the car will naturally reduce its speed to a standstill, *i.e.*, decelerate, this rate of speed reduction is by no means fast enough, so that additional friction creating devices, known as brakes, become a necessity.

Looked at from basic principles, a car moving at a given speed V feet per second has a definite kinetic energy, which is given by the relation

$$K.E. = \frac{M \cdot V^2}{2g} \text{ foot pounds}$$

where M is the weight of the car in lbs., and g is a constant of value 32·12.

Now, in order to bring the car to rest in a given distance d feet, it is necessary to provide a frictional resistance R lbs., such that

$$R \cdot d \qquad 2g$$

As an example, a car of weight 1 ton (2,240 lbs.), moving at 20½ m.p.h. (=30 feet per second), has a kinetic energy given by

$$\frac{2240 \times 30^2}{2 \times 32{\cdot}12}$$

Now, to bring the car to rest in 60 feet, the average resistance which must be exerted is given by the relation

$$R \times 60 = 31{,}350$$
$$\text{whence } R = 522{\cdot}5 \text{ lbs.}$$

The friction in the mechanism of the car, and the drag of the tyres, supply part of this resistance, and the brakes the rest.

In the best designs of brakes, it is arranged that the

foot brake, for rear wheel brakes only, shall bring the car to rest, from a speed of 30 m.p.h., in a distance of about 18 to 25 yards. With brakes on all four wheels, the foot pedal application should easily bring the car to a standstill in 8 to 10 yards from this speed. The time taken in the latter case is between 2 and 3 seconds.

The brakes convert the kinetic energy of the car into heat energy by means of the friction created between the brake linings—which are usually of asbestos fabric impregnated, and often reinforced with fine brass wires, as with Ferodo brake linings—and the metal surfaces of the drums upon which they act. That heat is generated when the brakes are applied may be demonstrated by jacking up one wheel and running the one wheel for a time with the brake on; the drum will be found to get quite warm. It is, therefore, necessary in well designed brakes to get rid of this braking heat energy as rapidly as possible; this is usually accomplished by providing cooling ribs on the outside surfaces of the brake-drums, and by making the latter as large as possible so as to reduce the heat dissipated for each unit of surface.

Braking Systems.—Braking systems which have been employed may be classified as follows:—(1) foot brakes on rear wheel drums; hand brake also on rear wheel drums. (2) foot brake on countershaft, *i.e.*, a drum on the gear-box final drive shaft extension, just in front of propeller shaft; hand brake on rear wheel drums. (3) foot brake on all four wheels; hand brake on all four wheels, or on rear wheels only.

The *foot brake* is operated by a pedal, which is usually placed in the centre between the clutch and accelerator pedals, and is used for normal driving purposes. The hand-operated brake is brought into action by a hand lever, with trigger ratchet and toothed quadrant, situated close to the gear change lever. It is intended to hold the car, when at rest, or to assist the foot pedal brake when descending long hills. The toothed quadrant mechanism holds the brake 'on'; it is disengaged by depressing the brake lever trigger.

Rear Wheel Brakes.—Pressed or cast steel drums, varying in diameter according to the size of car, from 10

inches to 18 inches, are fitted securely to the rear wheels. The *brake shoes*, as they are termed, usually consist of two metal pieces of segmental shape, hinged at one end, and held together by means of a spring at their opposite

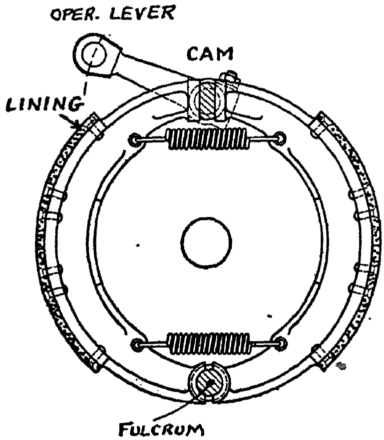


FIG. 233.—Principle of Internal Expanding Brake.

ends. A small cam-shaped piece is inserted between the latter ends, so that when the cam is rotated it forces the two ends apart, and causes the friction surfaces (riveted to the brake shoes) to grip the internal cylindrical surface of the brake drum. The principle of the *Internal Expanding Brake*—as this type is termed—is illustrated in Fig. 233. It will be observed that the two shoes are held 'away' from the drum by means of the *springs* shown; the brake operating lever, in rocking, moves the cam, between the shoes and expands them against the spring action.

Servo Type Brakes.—There is also another interesting design of brake in which there are two shoes, one large and one small, pivoted one to the other at one end. The other end of the larger shoe is pivoted to a fixed anchorage, whilst the other end of the smaller shoe is hinged to the brake shoe expanding device. When the latter is operated by the usual brake mechanism, it makes contact with the drum, and, being free, within certain limits, to rotate, is

dragged around, a little ; this brings the larger shoe into contact with the drum, and gives a much more powerful

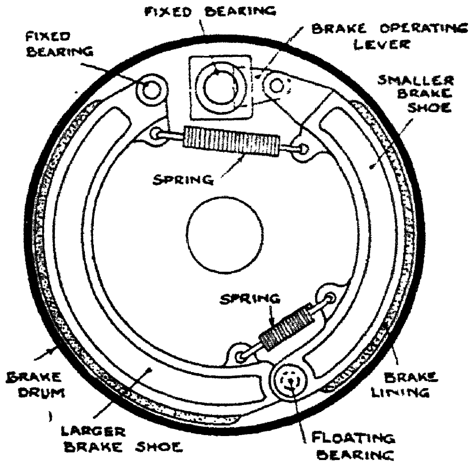


FIG. 234.—Floating Pivot, Servo Type of Brake.

braking effect ; this '*floating pivot*' type is really a kind of Servo system for augmenting the braking effect ; it utilises the energy in the wheels themselves.

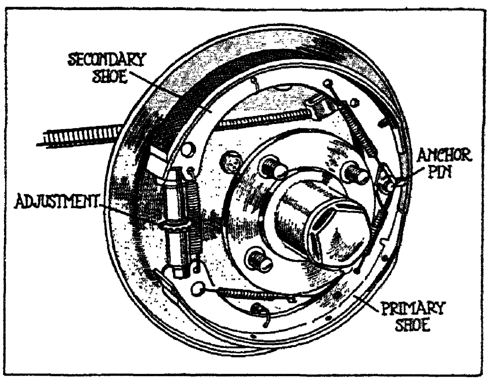


FIG. 235.—Armstrong Siddeley Brake.

Fig. 235 shows the Duo-Servo type of brake which is fitted to each of the wheels in the case of the Armstrong Siddeley cars. This belongs to the same class as that

illustrated in Fig. 234 and it comprises a primary and secondary shoe. The primary shoe is operated positively by means of the cable connection and lever shown; it will be noticed that this cable passes from outside to inside the back plate.

The other ends of the brake shoes, i.e., on the opposite side to the anchor pin are held in their 'Off' position by means of a link having a right and left-handed screw adjustment, for the purpose of taking up brake lining wear effects.

The adjustment of this type of brake is effected by removing a metal plate cover from a slotted hole in the back plate and inserting a screwdriver so as to turn the notched adjusting wheel. When all four brakes have been adjusted so as to be just clear of their drums—with the four wheels jacked up for this purpose—the brakes are equalised by pushing the brake pedal down and holding it there until the tightest wheel can just be turned by hand. The other three wheel brakes are then tightened to a similar extent. The secondary shoe has an eccentric stop which requires adjustment, about once every 15,000 miles.

The Girling Brake.—Instead of employing the usual rocking cam method of expanding the brake shoes the Girling system employs a wedge movement for this purpose, actuated by a pull-rod working at right angles to the plane of the brake drum. Referring to Fig. 236*A* there is a cone 1, which is pulled upwards by means of the rod 2; the latter is operated by the foot-brake mechanism. This movement forces apart the plungers 3, the outer ends of which engage with and expand the brake shoes; the rollers 4 serve to reduce the friction between 1 and 3. The housing 5 is held lightly on the brake back plate 6 by nuts and spring washers 7 so that it floats between the brake shoes, which are thereby rendered self-centring.

In order to take up the wear effects of the brake shoe friction material an adjustment unit is provided (Fig. 236*D*) on the opposite side of the back plate to the expander unit described; this adjuster is shown in Fig. 236*B* and consists of a screwed member provided with a squared end outside the back plate and a conical end with eight flats on it, on the inside of the back plate. As the adjuster is screwed down the conical portion *A* pushes the plungers

C outwards and as the ends of the brake shoes are held in slots in the outer ends of *C* the shoes are forced outwards to take up the brake wear. The complete adjusting unit is mounted in the housing *B* which is bolted to the back-

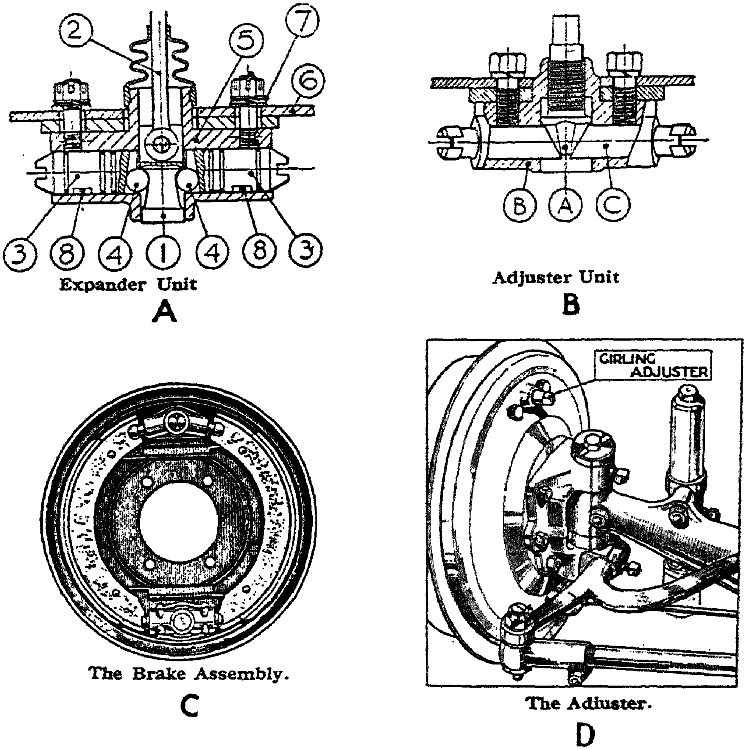


FIG. 236.—The Girling Brake (Austin).

plate, as shown. These brakes can be adjusted without jacking up the car wheels, by first turning the screwed members *A* downwards as far as they will go and then releasing them by about one full notch to give the necessary drum clearance.

Two alternative methods of operating Girling brakes are illustrated in Fig. 237. Referring to the upper illustration the front brake drums are shown at *A* and *B*. A three-armed lever *L* with fixed bearing pin is actuated by the tie-rod *C* from the foot brake pedal in an anti-clockwise direction, so that the two smaller arms of *L* provide the

necessary pulls on the tension rod members *P* and *Q* to actuate the brakes.

This method can also be applied to the rear brakes but a more simple system is shown in Fig. 237 (lower illustration).

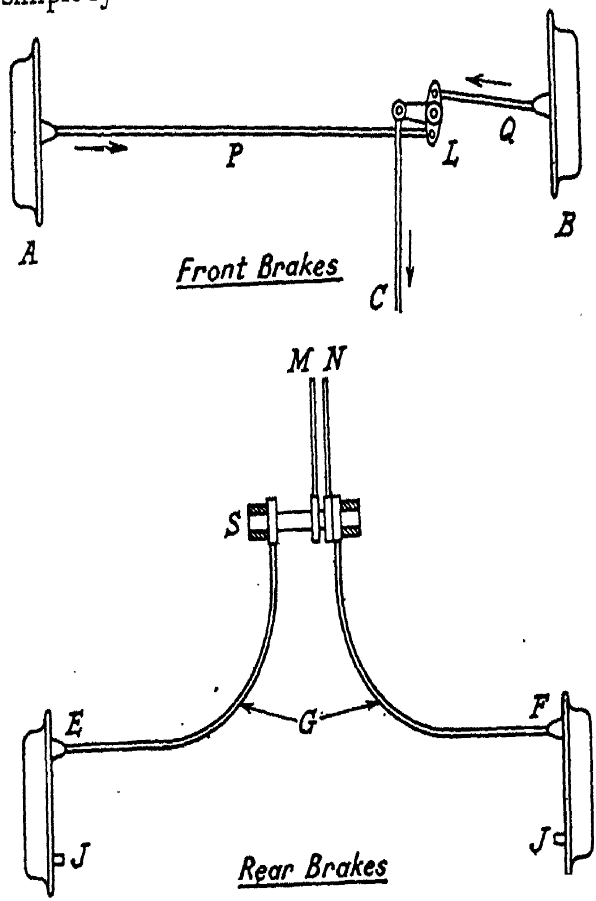


FIG. 237.—Methods of Operating Girling Brakes.

The Bendix Cowdrey Brake.—A more recent development in connection with the operation of these brakes is the use of the wedge expanding method on the principle of that employed in the Girling brake, but of different design. Referring to Fig. 238 the brake rod has a plunger head

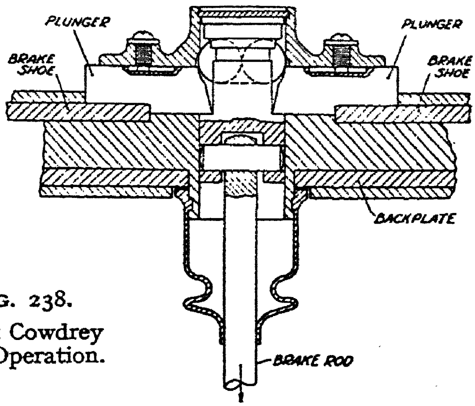


FIG. 238.
Bendix Cowdrey Brake Operation.

which engages with two steel balls located between the inclined faces of the plungers actuating the ends of the two brake shoes. One or other of the latter abuts against the housing—according to the direction of rotation of the brake drum. As the brake rod is pulled in the direction shown by the arrow the steel balls force the plungers outwards and apply the brakes; the shoes employed in this design of brake are of the self-energising type.

Bendix Auto Control for Brakes.—A clever scheme for brake compensation whereby the proportion of braking between the front and rear wheels is altered, automatically, is shown, diagrammatically, in Fig. 239. It is interposed in the brake rod actuating the rear brakes and is placed behind the main brake cross shaft and mounted on the chassis frame. The device consists of a cylinder C, rigidly secured to the frame, containing a sleeve member B and an internal wedge member A; these two members form part of the rear brake rod system. Normally the spring shown holds A against B leaving the rollers quite free. Thus, whilst the pull at A is less than the spring pressure the whole unit is free to slide in the housing C as a rigid

unit or single brake rod. When the brakes are ap. progressively the members A and B move as a single unit, but as soon as the maximum braking effort occurs at the rear wheels the spring compresses and the wedge

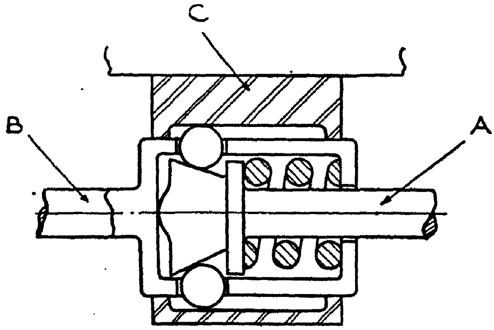


FIG. 239.
Bendix Auto Control Braking Scheme.

portion of A forces the rollers outwards against the inside of the rigidly mounted unit C, thus locking the rear brake rod and preventing any further movement. After this stage is reached, any further braking effort is transferred, automatically, to the front brakes thus preventing the rear brakes from locking.

Transmission Brake.—The transmission brake was once widely employed on motor cars. It had several points to recommend it. In the first place it was very powerful, since the back axle reduction gear actually multiplied the braking torque about 4 or 5 times. Secondly, it distributed the braking effort equally to the two wheels. Thirdly, it reduced the unsprung weight on the rear wheels, by dispensing with one set of brakes. Its disadvantages lie in the increased heating of the brake drum due to the greater braking effort, and to the fact that it is difficult to dissipate the heat owing to its enclosed situation. Further, it imposes greater strain on the transmission than the rear wheel type, since the braking torque is transmitted through the propeller shaft, differential and axle shafts. One result of this was to cause noise, whenever there was any backlash due to wear or bad adjustment, in the transmission; this rattle is particularly noticeable in some cars.

Although discarded in modern cars for foot-braking

purposes the transmission brake is still occasionally employed for commercial vehicles and cars for the hand-braking system; thus on certain Chrysler and other American cars the hand-brake actuates a contracting brake band on a drum mounted on the rear end of the gear shaft.

Four Wheel Brakes.—With the general all-round improvement in the speed and acceleration of modern cars, there was felt the need for a corresponding improvement in the braking system. Although the transmission and rear wheel brakes had been developed to a high standard of efficiency, by the use of larger brake drums, better frictional material and more efficient designs of brakes, and these in the hands of a good driver provide good deceleration means for all ordinary purposes, a limit to the stopping power had been reached. As there is an increasing tendency to drive at higher road speeds, so there has also arisen a demand for more powerful braking for safety reasons. Rear wheel brakes are effective only so long as the rear wheels do not skid, or lock. It is quite possible to design transmission and rear brakes which will lock the wheels, so that the latter slide instead of rotating when full pressure is applied to the foot brake. This limit to the braking depends upon the weight on the rear wheels, the nature of the road surface and upon the type of tyre used; the greater the weight and the more 'non-skid' the tyres, the greater will be the braking effort before skidding. This limitation is due to the adhesive force of the rear wheels, as explained in Chapter X.

Having reached the limit of braking on the rear wheels, it is a natural consequence that adhesive force of the front wheels should be pressed into service. In the ordinary way from one-third to two-fifths of the car's weight is borne by the front wheels, so that the braking effort when all four wheels are used will be increased accordingly.

Braking Efficiency and Stopping Distances.—A practical measure of the efficiency of the braking system is that of the minimum distance in which the car, after travelling at a given speed can be brought to rest when the foot-brake pedal is applied. The distance in question will

depend upon the adhesion between the tyres and the road surface; this, in turn concerns the condition of the tyre tread and the inflation pressure, the road surface nature and similar factors.

Tests have shown that the resistance to sliding experienced by a car on the road may be equal to or even greater than the total weight of the car when braked violently, although the results of towing tests show that the adhesion seldom exceeds 70 to 75 per cent. of the total weight. The results of tests made with modern braking systems indicates that the stopping distances can be expressed by the following relation :—

$$D = k.V^2$$

where D=stopping distance in feet, V=velocity in miles per hour and k a constant, depending upon the state of the tyres, inflation pressure, nature of the road surface, etc.

The value of k for a well-designed braking system and average dry concrete roads is approximately $\frac{1}{25}$ for four wheel brakes; incidentally it has a value of about $\frac{1}{12}$ for the earlier two-wheel braking systems. The stopping distances will also depend upon the condition or adjustment of the brakes and in this connection the following table, based upon practical road test results* may be found useful :—

Stopping Distances and Brake Condition

Speed in M.P.H.	Condition of Brakes			
	Perfect	Good	Poor	Dangerous
	Stopping Distances in Feet from			
20	15	19	28	33
30	34	43	63	75
40	60	76	112	132
50	95	120	176	210
60	136	172	250	300

The braking deceleration† of a vehicle is sometir

* Raybestos-Belaco Ltd.

† Brake efficiency calculations are given in Chapter X.

expressed as a percentage of the acceleration 'g' due to gravity, namely 32·2 ft. (or nearly 22 m.p.h.) per sec., per sec. A braking system capable of giving the same deceleration as gravity is said to have an efficiency of 100 per cent. Modern braking systems are so good, however, that decelerations of 20 to 24 m.p.h. per sec. are possible, so that efficiencies—on this basis—of over 100 per cent. are possible.

Effect of Road Surface Condition. The nature and condition of the road surface has an important influence upon the braking effect, different surfaces giving different stopping distances. In this connection the results of tests made by the Ferodo Company show that, whereas, for *rough concrete roads*, a car fitted with four wheel brakes can be brought to rest from 30 M.P.H. in 30 feet, for *smooth concrete* surfaces this distance becomes 33 ft.; for *dry macadam* it is 38 ft. and for *wet macadam*, 50 ft. For *dry and wet asphalt surfaces* the stopping distances are 43 and 60 ft., respectively.

The Hand Brake.—In the earlier two-wheel braking systems the foot pedal usually operated either a transmission brake mounted behind the gear-box or a pair of brake shoes, namely, one in each rear brake drum. In addition, there was fitted a hand-operated braking system which was independent—as the law required—of the foot brakes. Usually this was effected by means of separate brake shoes, one in each rear brake drum.

Nowadays, with the much more efficient four-wheel braking systems in use, the hand-brake is employed merely as a parking brake to hold the car whilst at rest on the level or on an incline. It is usual to connect up the hand brake operating mechanism to the rear brake shoes, so that the same shoes are employed for both the foot and hand brake. In some cases, however, a separate pair of shoes is employed in the rear brake drum for the hand-brake or a transmission hand-brake is provided; this results in two independent braking systems.

The hand-brake lever normally employed is placed on one side—usually the left-hand of the central gear change lever; but this has the drawback of occupying a certain amount of room in the passenger's compartment

and it is not very convenient for the driver to operate. It is an advantage, therefore, to arrange for the hand-brake lever to be mounted horizontally in the space between the two front seats, where it is not only more convenient to operate, but enables the driver to exert

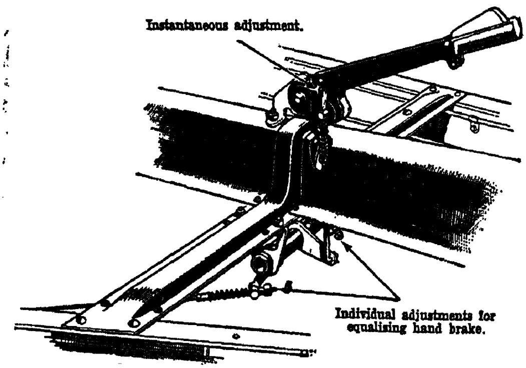


FIG. 240.—The Morris Hand Brake.

a direct and powerful braking effort ; moreover it does not interfere with the passenger's comfort. Pistol grip hand brake levers on the dash are also fitted.

Fig. 240 shows the Morris Eight hand brake which is arranged on these lines and is applied with an upward movement. It operates upon the rear brake shoes by means of cable mechanism and is independent of the hydraulic brakes. Both instantaneous and compensating adjustments are provided, the former being effected from the driver's seat.

A Modern Transmission Brake.—Reverting to the subject of transmission brakes a recent return to this method on the part of certain American manufacturers is shown in Fig. 241. It is known as the 'Detroit Duo-Grip' brake and has two segmental shoes located on the outside and inside of a portion of the drum, these shoes being brought into contact with the latter by means of two forks which are provided with adjustments for taking up the wear. Only one-third of the brake drum surface is engaged, the dual action being equivalent to more than one-half of the surface. Moreover, the rest of the brake

drum surface is exposed to the cooling action of the air; the brakes and drum are also cooled by the impeller action of slotted holes drilled through the brake drum web at an

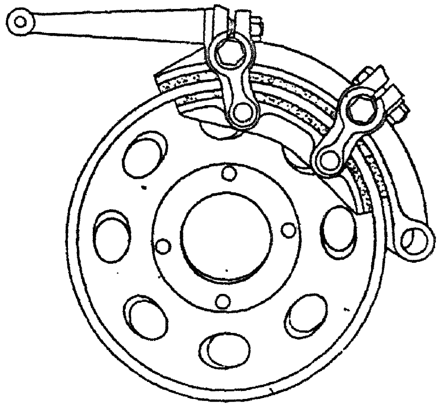


FIG. 241.
The Detroit Duo-Grip Transmission Brake.

angle. The action of the brake shoes is a balanced one so that there is no appreciable loading of the gear shaft bearings. A further advantage is the readiness with which the brake shoes can be detached for relining or replacement.

Types of Four Wheel Brakes.—There is a variety of designs in present use, some purely mechanical in action, some partly mechanical and partly hydraulic, or pneumatic.

In the case of small and medium cars, the mechanical type is preferred. For larger cars the engine-operated (or Servo), and the hydraulic and pneumatic systems are employed; there is also a vacuum system, similar to that used on railway trains, in use on some cars and commercial vehicles.

The Pneumatic Type.—This system, which is employed on commercial vehicles, has been used on one or two cars, but it has not proved popular. The general scheme adopted is to drive a two-stage air compressor from the engine crankshaft, or a gear-box shaft, the compressed air being delivered through a cleaning device to a reservoir tank under the body of the car; a relief valve is fitted

to prevent the pressure rising above a certain amount; when the relief valve operates, the compressor is automatically stopped. Each of the four wheel brakes is operated by a simple compressed air cylinder and piston, the latter being connected to the brake shoe cam operating lever.

Pressure on the foot brake pedal opens the control valve, and allows the air under pressure in the reservoir to act upon the four brake pistons. The system requires careful design in order to provide for the smooth application of the brakes when the control valve is opened. It is usual to fit an emergency mechanical brake to obviate the drawbacks associated with a failure of the air supply. The compressed air in the reservoir can be used for tyre inflation purposes.

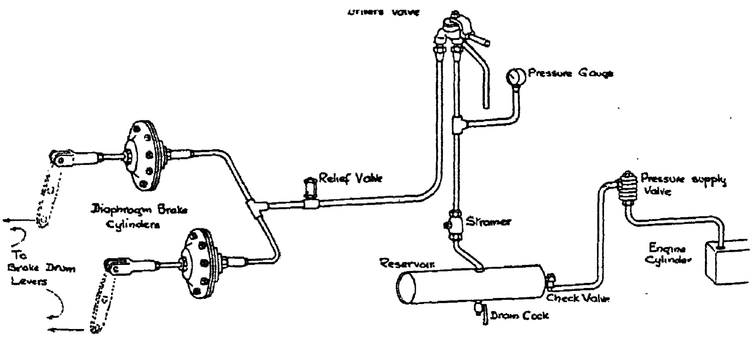


FIG. 242 —Westinghouse Pneumatic System for Automobile Braking.

In the case of the well-known Westinghouse air pressure system used on commercial vehicles, there is the usual air pump, reservoir and brake-pedal-operated control valve, but in this system the latter, when opened, admits air under pressure to a diaphragm chamber (Fig. 242), whence it deflects a circular steel diaphragm clamped, so as to be air-tight, around its edges. The deflection of this diaphragm actuates the brake mechanism, a rod from the former connecting it with the ordinary brake rocking shaft mechanism. As the movement is small, the mechanism must be designed to multiply it at the brake drums. This system is applicable to front as well as rear wheels, and also to the wheels of commercial trailers;

in effect, it really comes under the heading of *pneumatic servo brakes*, since it increases the effort of the driver on the brake pedal.

The Hydraulic Type.—A common advantage of both pneumatic and hydraulic types of four wheel brakes over purely mechanical ones, is that equal braking efforts can always be obtained at all four wheels, or on pairs of wheels, due to the existence of equal air or fluid pressures.

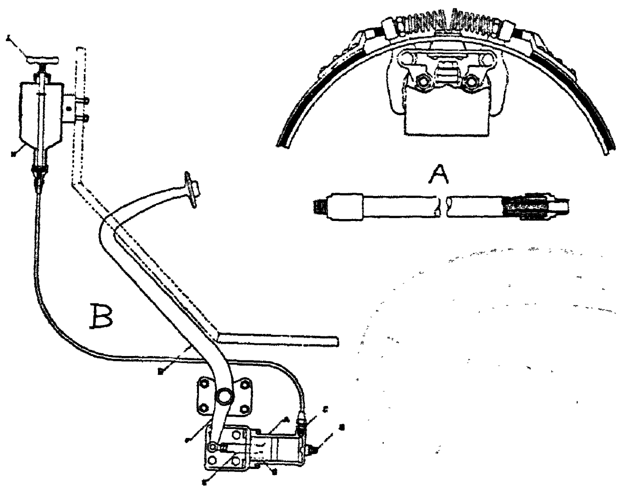


FIG. 243.—The Original Lockheed Hydraulic System of Braking. *A*. Brake Mechanism. *B*. Brake Pedal Piston and Oil Reservoir for Replenishing the System.

Another big advantage is the increased braking effort obtained, thus obviating the need for Servo motors; the complicated mechanism (working joints and rods of the mechanical types) is also avoided, so that simplification results.

The principle of the system mentioned is to employ oil under pressure and to transmit this pressure through pipes (some being flexible) to the oil-pressure operated pistons working in cylinders in the brake drums of each of the four wheels. Sometimes a separate oil pump is employed for creating the pressure. In others, the act of depressing the brake-pedal forces a piston into a cylinder of oil, and creates the necessary pressure in the system

to supply the four brake cylinders and pistons. In the earlier Lockheed type (Fig. 243), the brake-pedal operated a small piston, and supplied pressure through suitable metal tubing to each of the four brake cylinders. In each of the latter there were two pistons, which, under the oil pressure between, were forced outwards, thus causing pivoted bell-crank levers to rock, in such a way that their outer ends moved together; this movement drew the two ends of an external band brake together and thus applied the brakes. The only connection with the chassis in this case was a small flexible oil pipe to each brake drum.

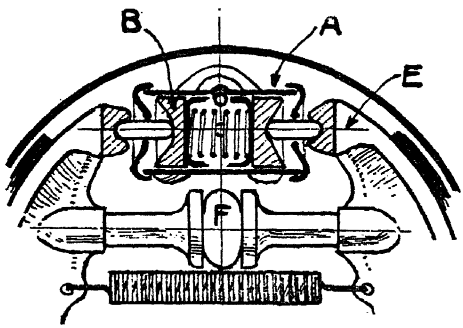


FIG. 244.—Lockheed Wheel Cylinder Arrangement.

The Internal Lockheed Brake.—The later type of Lockheed brake is of the internal expansion brake shoe type, the whole of the brake mechanism being contained inside the brake drum; here it is protected against dirt and moisture.

The principle of the internal shoe type hydraulic brake is shown in Fig. 244. When the foot pedal is depressed oil pressure is communicated to the space between the pistons *B* in the wheel brake cylinder *A* and the pistons are thereby forced outwards. The short struts between the outer sides of the pistons and the ends of the brake shoes *E* communicate this movement to the latter and press them against the brake drum. In this particular design there is a brake cam *F* which can be operated, in the usual manner, by means of a rocking shaft so as to apply the brake shoes independently by mechanical action only; this method is intended for hand-brake operation of the rear brake shoes.

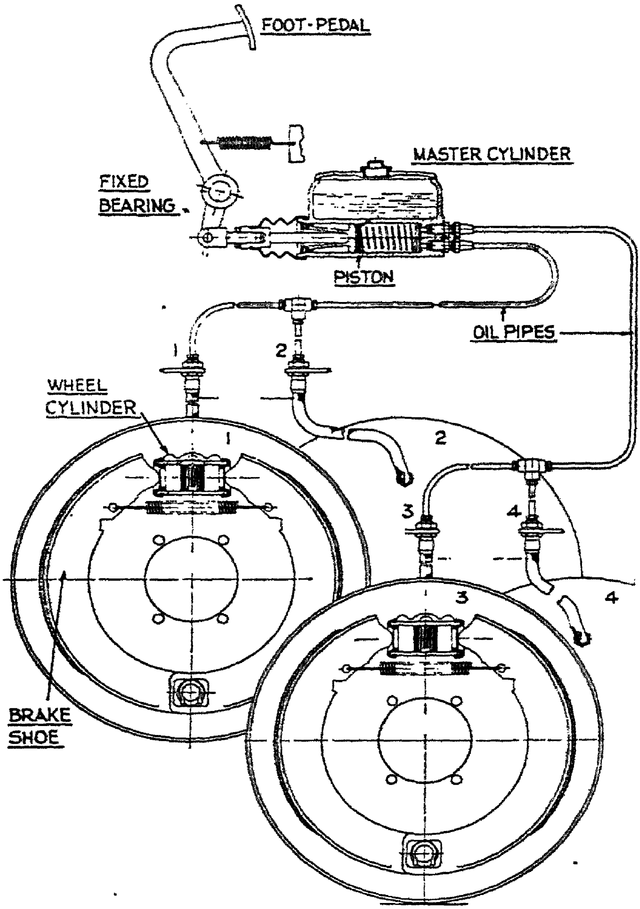


FIG. 245.—Lockheed Hydraulic Braking System.

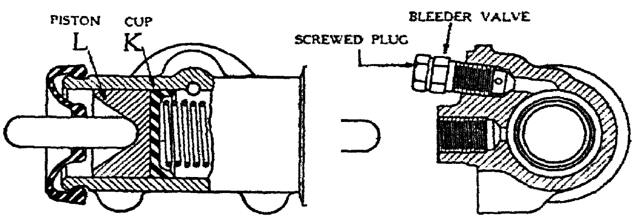


FIG. 246.—Brake-Operating Pistons and Bleeder Valve.

One of the pairs of oil-operated pistons and its cylinder is shown in part section in Fig. 246. The piston *L* is made of metal and is provided with a cup-leather *K*; there are two such pistons and cup-leathers in each brake cylinder. The projections at the extreme ends of the pistons engage with the tops of the brake shoes. On the top and in the centre of the wheel-cylinder a special priming or 'bleeder' valve is provided; this is used when filling the system with oil, to get rid of the air in the system.

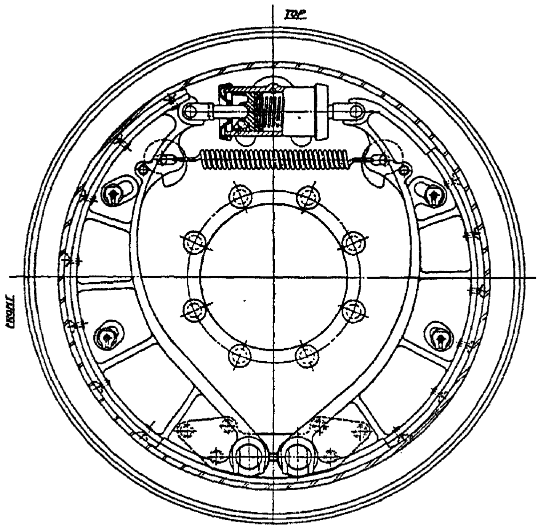


FIG. 247.—Modern Hydraulic Brake Drum Unit.

Fig. 247 shows the arrangement of the internal brake, the oil cylinder, brake shoes, pivot pins and shoe return spring being shown clearly.

In the more recent hydraulic brakes a compensating type master cylinder is arranged to maintain a constant volume of oil in the system when at rest at a uniform pressure of 8 lb. per sq. inch, this pressure acting as a liquid expander on all rubber cups, ensuring complete and efficient sealing of the system. There is no loss of liquid due to evaporation, as the fluid chamber is airtight, but automatic compensation is provided for expansion or contraction of the fluid due to temperature changes by inlet and outlet valves. The special fluid used in the system

is immune from freezing and unaffected by high temperatures.

The master cylinder (*M*, Fig. 248) is contained within the supply tank upon which is mounted the pedal; the whole unit in turn is mounted directly on a convenient part of the chassis. This supply tank serves to carry the reserve

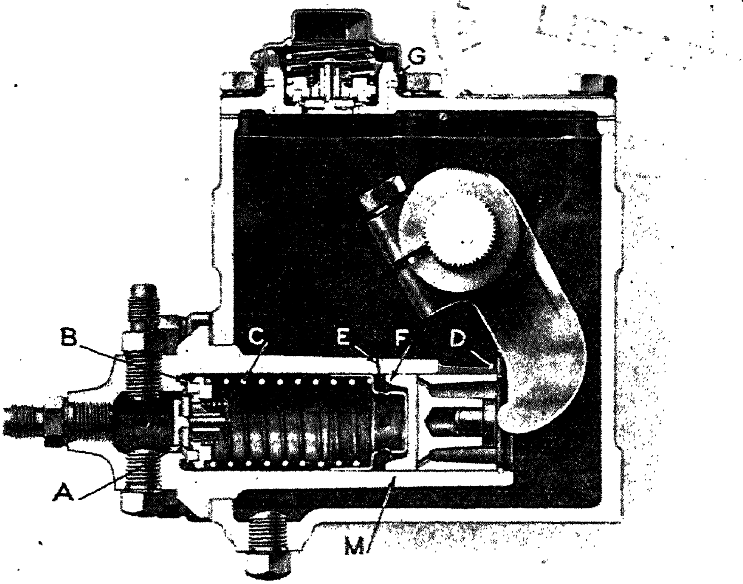


FIG. 248.—The Master Cylinder.

supply of fluid. The master cylinder is submerged in the fluid, thereby preventing the system from taking in air, dirt or water.

Located so as to be directly forward of the master piston cup, when piston is in the 'off' position, is a port-hole through the wall of the cylinder (*E*, Fig. 248).

Any rise in temperature causing the fluid in the braking system to expand allows the fluid to pass through the port to the supply tank. Any drop in temperature causing the fluid to contract allows the fluid to flow back through the port. Thus a constant volume of fluid is maintained in the system.

The brake-pedal must, of course, be set so that the master

K*

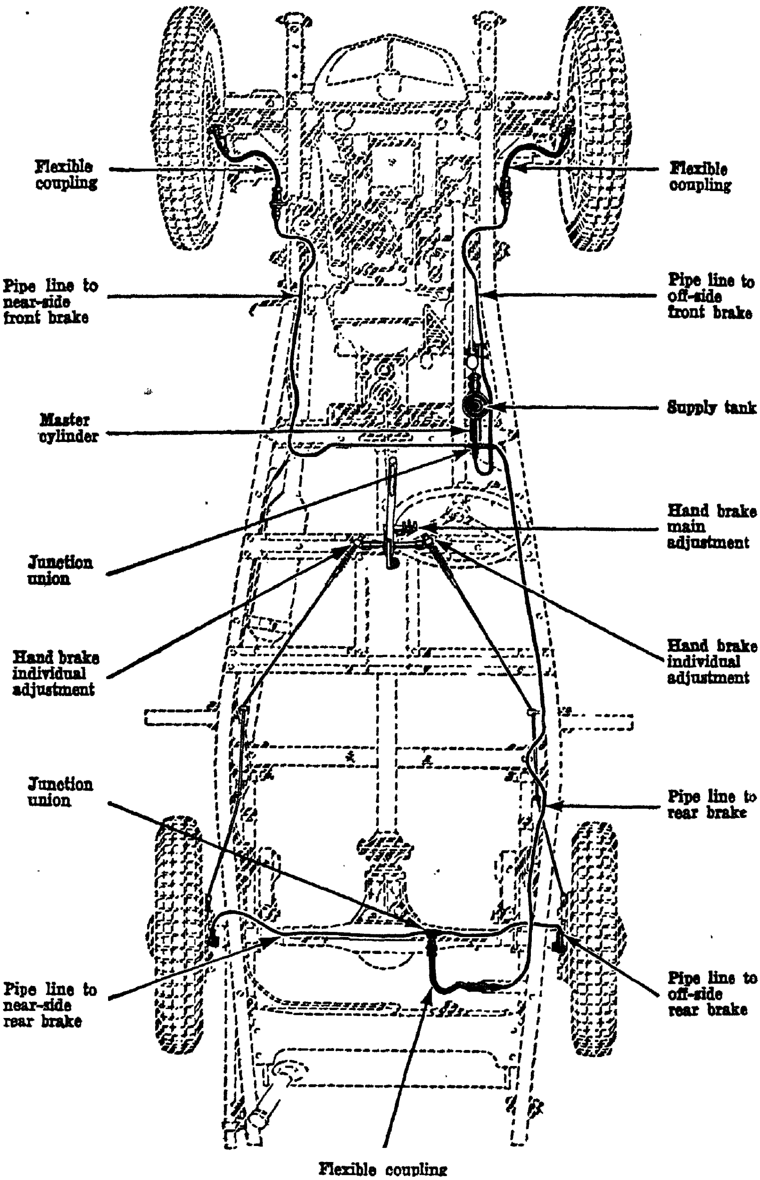


FIG. 249.—Morris Hydraulic Braking System Layout.

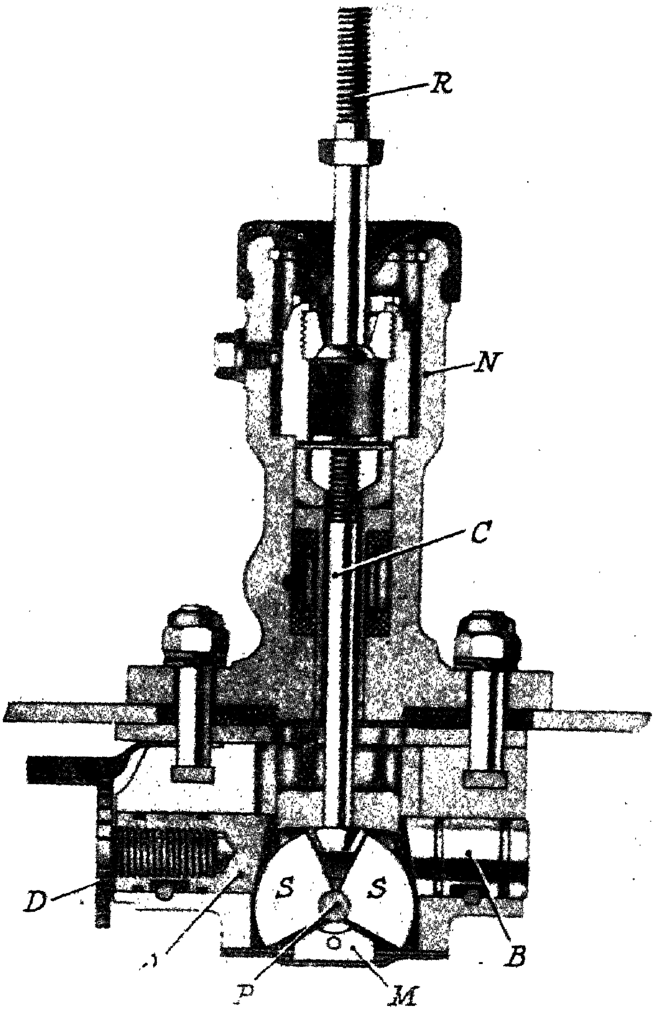


FIG. 250.—Lockheed External Cylinder Hydraulic Brake Unit.

piston cup is back or clear of the port when brakes are in the 'off' position.

The return of the master cylinder piston to its 'off' position is accomplished by a return spring (*C*, Fig. 248) enclosed within the master cylinder.

In the head of the master cylinder, held in place by a return spring, is a combination inlet and outlet check valve. When brakes are applied, the master cylinder piston is pushed forward, which opens the outlet check valve *A*. The fluid is forced through it into the system. When the brake foot-pedal is released, the master piston return spring forces the piston back to its 'off' position against its stop *D*. At the same time the wheel cylinder piston return springs are forcing the fluid back through the inlet check valve *B* until the fluid pressure balances the weight of the master cylinder return spring, when the inlet valve closes.

External Cylinder Hydraulic Brake. An improved design of hydraulic brake, known as the Lockheed Bisector Expander (Fig. 250) employs a single external oil cylinder instead of the double piston internal one previously described. The cylinder *N* which is mounted on the back plate has a piston attached by means of a central rod *C* to a lower member *M* having two sectors *S* that can rock about a central pin *P*. These sectors engage with the two plungers *A* and *B* which then move outwards and since their outer ends are connected to the ends of the brake shoes the latter are forced outwards against the inside of the brake drum.

In addition there is an external rod *R* connected to the piston for operating the member *M* mechanically; this rod is actuated by the hand brake mechanism for independent braking such as parking purposes or for emergency should the oil system of the hydraulic brakes fail. A single screw adjustment *D* is provided to one plunger, namely, *A*, to enable wear of the shoes to be taken up. This adjustment is accessible by a screw-driver through a slot in the back plate of the brake unit; the assembly is made of the self-centring or 'floating' type, for this purpose.

The Tandem Master Cylinder. This later development of the Lockheed system is employed upon larger cars and commercial vehicles. It provides separate oil outlets for the front and rear braking systems so that in the event of a failure in one of these the other is unaffected. There are two separate circuits of oil under pressure and two

pistons in the master cylinder—which is rather larger than usual.

Referring to Fig. 251 one piston is connected to the foot

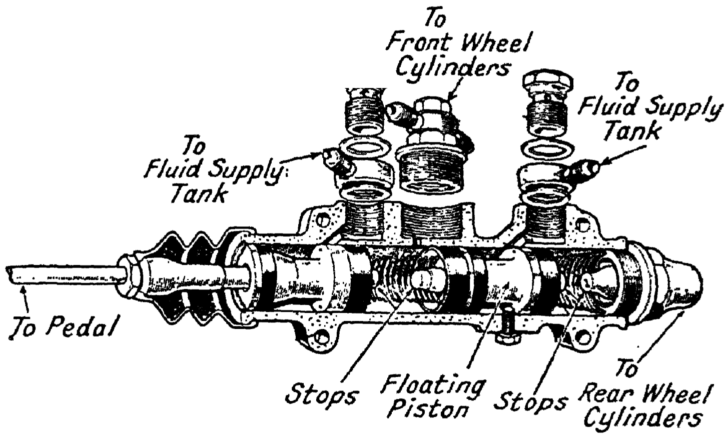


FIG. 251.—Lockheed Tandem Master Cylinder Unit.

brake pedal, whilst the other is a 'floating' piston in the middle with inlet and outlet pipes on each side. When the brake pedal is depressed pressure is created in the forward chamber and owing to the fact that the central

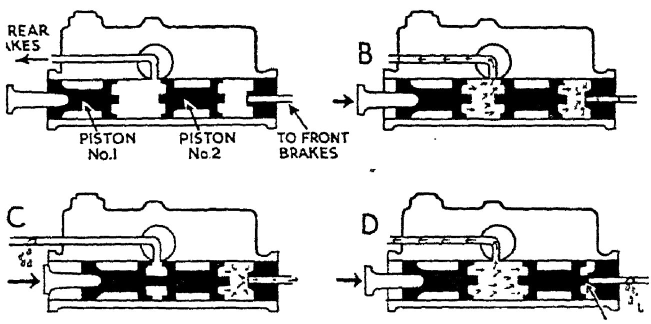


FIG. 252.—Illustrating Operation of the Tandem Master Cylinder System.

piston is not fixed pressure is also produced in the rear chamber so that in this way all four brakes are applied.

The operation of this system is shown in more detail in Fig. 252 in which Diagram *A* shows the general layout. In Diagram *B* the brakes are shown in normal operation,

the foot brake pedal forcing No. 1 piston along the master cylinder, moving No. 2 piston along as previously explained.

Should a leak occur in the piping leading to the rear brakes, as indicated in Diagram C, the No. 1 piston slides along so as to make contact with No. 2 piston and then transmits pressure to the front brakes.

If, however, a leak should develop in the pipe line leading to the front brakes then piston No. 1 transmits hydraulic pressure to the rear brakes and No. 2 piston slides along to form a buffer against a stop at the end of the cylinder.

Girling Hydraulic Brake. The mechanical method of operation of the Girling brake described earlier in this chapter can be replaced by an hydraulic system, whereby the rods and rocking levers are dispensed with and oil pipes connected to cylinders on the brake back plates employed

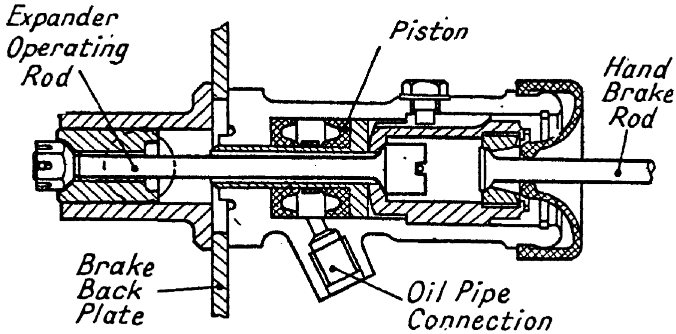


FIG. 253.—Girling Hydraulic Brake.

instead. Each brake cylinder has a single piston which actuates the brake expander direct. In other respects the design of the brake unit is the same as for the mechanical type.

Fig. 253 shows a typical hydraulic connection unit for the brake back plate. The central rod is moved to the right when oil pressure is applied to the piston. The hand brake connection shown enables the brake to be applied mechanically for parking or hill-holding purposes, or for use in the rare event of a failure of the oil system.

Typical Hydraulic Braking System.—The layout of a typical hydraulic braking system as used on Morris cars

is shown in Fig. 249, the various components being indicated by the arrows. In the smaller Morris models the master cylinder is combined with the oil supply tank which is mounted beneath the foot-board; the supply tank is simply a reservoir to supply oil to feed the braking system over relatively long periods.

Taking Up Brake Shoe Wear.—When the linings of the shoes in the Lockheed hydraulic braking system become worn it is necessary to take up the clearances between the worn friction material and the brake drum, so that the foot pedal will continue to exert its full braking movement, without touching the floor boards. To do this it is necessary to rotate by a small amount each of two hexagon headed

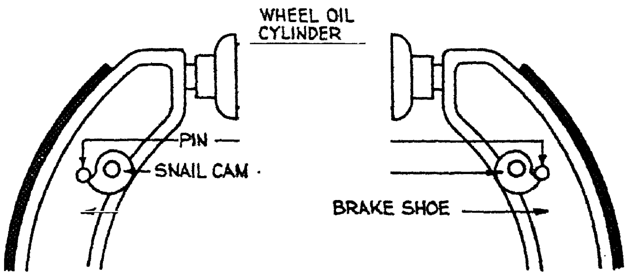


FIG. 254.—Method of Adjusting the Brake Shoes.

adjusting bolts. These operate snail-cams (Figs. 247 and 254) which bear upon pins in the brake shoes and enable the latter to be adjusted in relation to the brake drums. It is necessary to rotate the bolts away from the centre of the wheel in order to take up the effects of brake lining wear.

The later model hydraulic brakes have a much simpler method of adjustment, namely, by means of screwed connectors between each piston and its brake shoe whereby the distance between the piston and shoe can be altered by screwing this connecting member into or out of its piston. The screwed connector is provided with a serrated wheel for this purpose and it is therefore necessary only to put a screw-driver through a hole in the brake drum—which is turned until the hole is opposite the serrated wheel—and then to rotate the latter by means of the

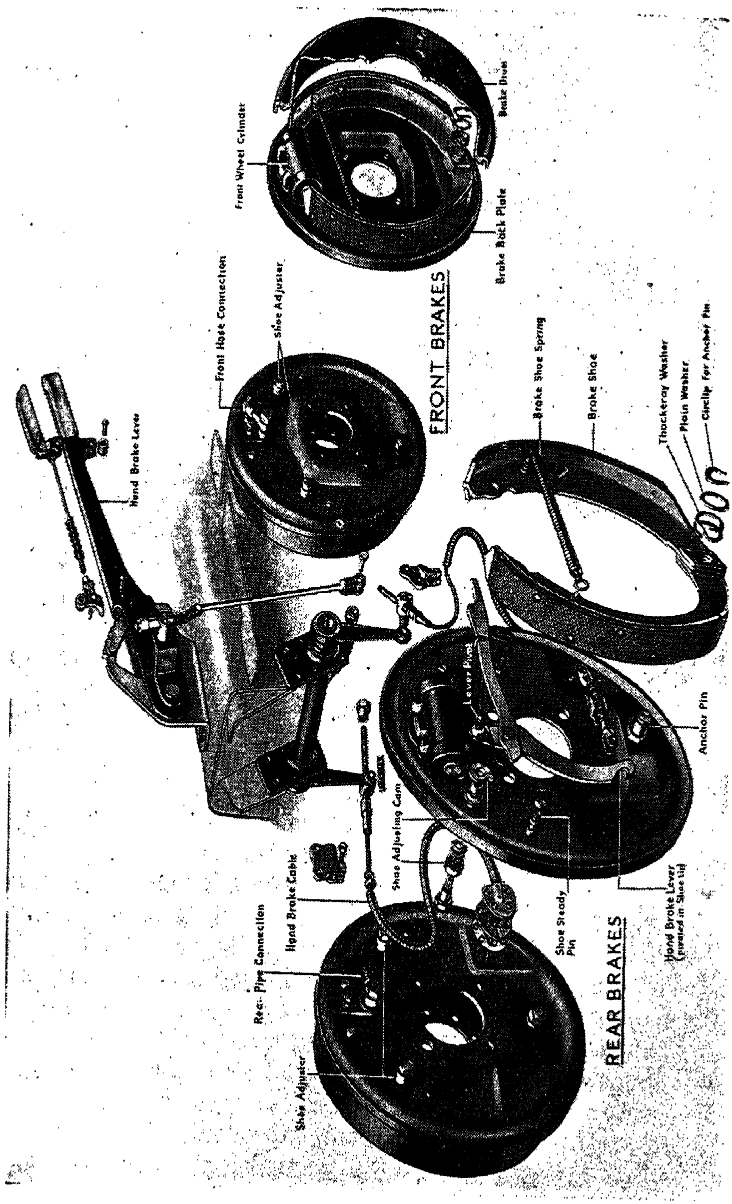


FIG. 255.—Complete Hand and Foot Lockheed Hydraulic Braking System of Later Design.

screw-driver blade. To take up wear the wheel is jacked up and the serrated wheel screwed towards the shoe until the wheel binds. It is then slackened back four notches. This operation must be done to each of the two adjusters on the brake. In another more recent design (Fig. 256) a single adjusting serrated wheel is employed. Access to this is through a hole in the brake drum. The hand

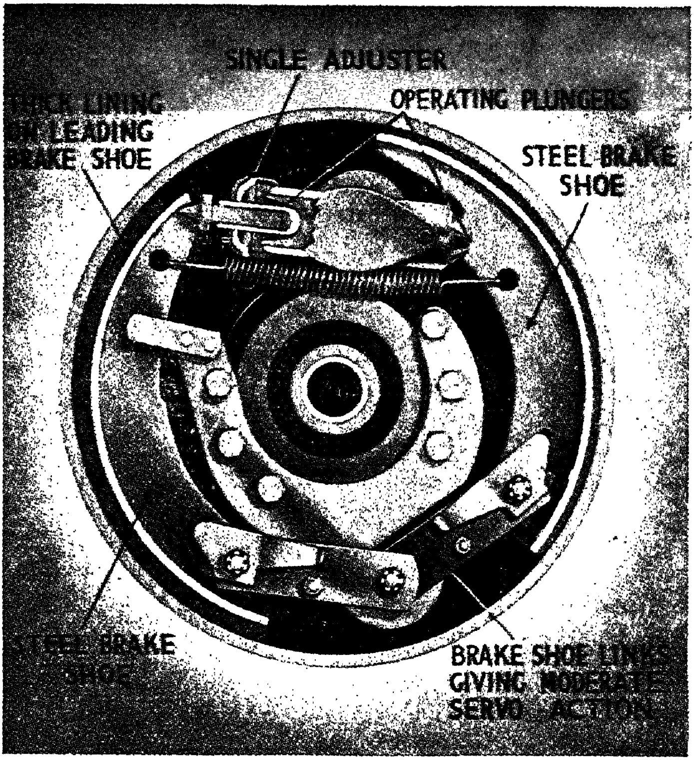


FIG. 256.—Single Adjuster Type Hydraulic Brake.

brake, in the later designs, operates the brakes of all four wheels, mechanically. The leading brake shoe *has a thicker friction lining* than the other shoe to take account of the increased amount of wear on this shoe.

Effect of Oil on Brakes.—Should any oil or grease get on to the brake shoes of one of the wheel drums, the

effect will be to reduce appreciably the braking torque on this wheel so that when the brake foot pedal is depressed the car will tend to skid; if the car is travelling fast, the effects may be serious. The cause of the trouble should be ascertained and remedied and the oil or grease removed from the surfaces with petrol, using a wire brush to roughen the friction material, afterwards.

Care of Hydraulic Brakes.—*In the event of an oil pipe breaking*, on the single master cylinder type, pressure would be lost, and the system would fail, or operate inefficiently. The hand brake is usually of the mechanical type, and can, therefore, be used in emergencies and to hold the car when 'at rest.' (Fig. 255).

It is essential that the glands of the operating piston rods be kept oil tight always, by screwing up when signs of leakage are apparent.

Special oils are used for these hydraulic systems; they are not too viscous, and possess a very low freezing point. In one well-known system, a mixture of glycerine and alcohol is employed. It is necessary in the earlier pattern Lockheed system to provide a supplementary reservoir containing oil, and in connection with the master cylinder or valve chamber, for the purpose of making up for any small leakages of oil.

The Vacuum Brake System.—In this system the power required to operate the brakes is derived from the partial vacuum existing in the inlet manifold when the engine is running. This vacuum is applied to one side of a fairly large piston working in a cylinder, the other side being exposed to the pressure of the atmosphere; it is the difference between the latter and the pressure in the inlet pipe which causes the piston to move, and, through suitable mechanism, to apply the brakes.

This system requires only the small effort imparted by the driver's foot on the brake-pedal to apply a very much larger effort on the brake operating mechanism; it, therefore, belongs to the assisted, or *Servo* brake class.

In the Dewandre vacuum servo brake, which is fitted to several makes of car, including the Daimler, the reduced inlet pressure (equivalent to 14 to 18 inches of mercury, or about 7 lb. per sq. inch in extreme cases)

is caused to act on a relatively large diameter piston, which can move in a cylinder. The movement of the piston when the foot-brake controlled distributing valve is opened, is communicated to the usual brake-operating mechanism, a chain being employed, as shown in Fig. 257.

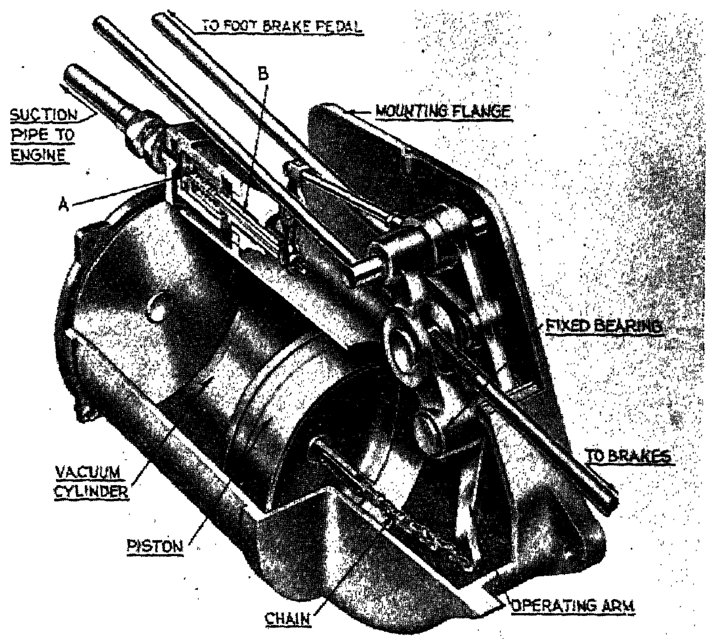


FIG. 257.—Dewandre Vacuum Brake Cylinder Unit.

There is also an ingenious device for taking the brakes off. The distributor contains two valves, one marked *A* for opening the inlet pipe to the cylinder of the brake piston, and the other marked *B* for placing the brake cylinder into direct connection with the air, for debraking purposes. The piston moves to the left, when there is a partial vacuum in its cylinder, and the chain moves the lever shown, about its pin; the brake-rods are operated by this lever. When the foot-brake pedal is released, the air-valve *B* is opened so that the brakes are released. With this system a progressive braking action is obtained, the braking force depending upon the amount of depression of the foot-pedal.

In order to preserve the usual conscious braking action of the driver, it is arranged that the pressure felt on the brake-pedal increases with the pull of the piston. The whole unit is enclosed in a dust-tight chamber, and it can be attached to any convenient part of the chassis frame, the only

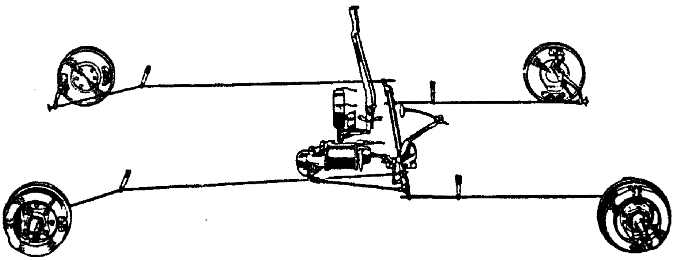


FIG. 258.—Vacuum Servo Brakes used on Small Cars.

addition being the attachment of the brake levers, and a pipe to the engine induction pipe.

The adjustment of this type of brake can be arranged—as in the Daimler system—by a single handle under the bonnet of the car; all four brakes are adjusted at once.

Front Brake Operation.—Front wheel brake design resembles closely that of the ordinary rear wheel brakes, in the matter of the brake shoe and brake drum arrangement. It differs, however, in the mode of operation, for the brake mechanism must operate equally well whether the front wheels are being steered or are running straight ahead; in other words it is necessary, from the fixed brake pedal system to operate the brake shoes whilst the wheels are in any of the steering positions. In order to accomplish this, the point or line of application of the brake mechanism for expanding the shoes must lie on the axis of the steering swivel pin produced. There are many methods of doing this, a popular method that was used when four-wheel brakes were first introduced being, to arrange for a rod sliding in a central hole through the king-pin to operate the brake cam through a suitable linkage.

In the Girling front brake arrangement the brake-operating pull-rod is arranged below the axle and it has a pin joint situated on the king-pin axis extension; in this way the steering does not effect the brakes in any way.

In the cable-operated Bendix Duo-Servo brakes that

have been widely employed on motor cars, the flexibility of the cables and their casings is such that no steering movement of the front wheels can affect them, so that the braking system is quite independent of the steering system.

With the hydraulic braking system also, the use of flexible oil pressure pipes between the chassis frame mounting and the front wheel brake drums obviates any interference between the steering and braking systems.

Servo Brakes.—In the case of the heavier makes of motor car, it is not, as a rule, possible for the driver to exert sufficient pressure on the foot brake pedal, to supply the necessary force required to operate the powerful brakes fitted. For this reason the braking equipment sometimes includes a mechanical device for supplying the power required to operate the brakes; all that the driver then has to do is to control this power, for braking purposes; this is analogous to the well-known electric relay devices.

The hydraulic and vacuum types of brakes described, have to a large extent displaced the purely mechanical Servo patterns. In the case of larger cars and many commercial vehicles, however, it is now usual to employ the vacuum type of brake to operate the piston device of the master cylinder of the hydraulic braking system, thus relieving the driver of this physical effort. A typical system is that known as the "Servo-assisted Lockheed Hydraulic Braking" one. The brake foot pedal then operates a piston valve which allows the reduced pressure or vacuum from the inlet manifold to communicate with one side of the vacuum cylinder piston; atmospheric pressure on the other side then applies the force needed to move the master cylinder piston.

The general layout of a vacuum-assisted hydraulic brake unit as used on a large car is illustrated in Fig. 259. The depression of the brake pedal downwards to the left causes the vacuum cylinder control valve rod to open a valve which places the right hand side of the vacuum cylinder into communication with the inlet manifold; so that the piston moves to the right and its rod—which is connected to the lower end of the lever *L*, hinged at *M* causes the end *N* of the push rod to the master cylinder to move to the left, thus operating the brakes.

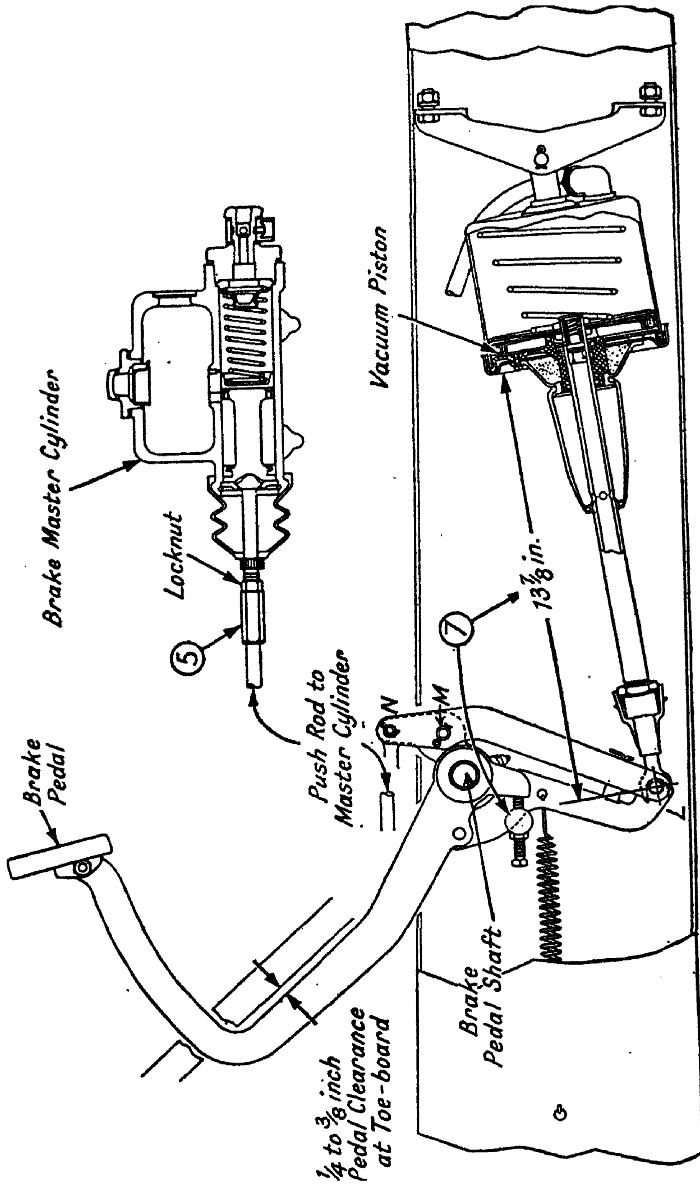


FIG. 259.—A Vacuum-assisted Hydraulic Braking System.

The upper view of the brake master cylinder is shown in reverse position, namely, as seen from the opposite side

of the chassis, so that the corresponding movement of the same push-rod 5 is to the right. The adjustment shown at 7 is for the correct relative positioning of the brake pedal.

The mechanical types of Servo devices are those in which the engine, or gearbox transmission supplies the necessary force; the act of depressing the foot brake pedal brings into engagement a clutch, which virtually couples up the engine or transmission driving shaft with the brake mechanism. It is proposed to describe one or two representative examples of Servo brakes; in this connection it should be remembered that oil-operated, or hydraulic, and also pnuematic and vacuum brakes all come into the category of Servo brakes, since the braking power is supplied, not by human effort, but by external sources of power.

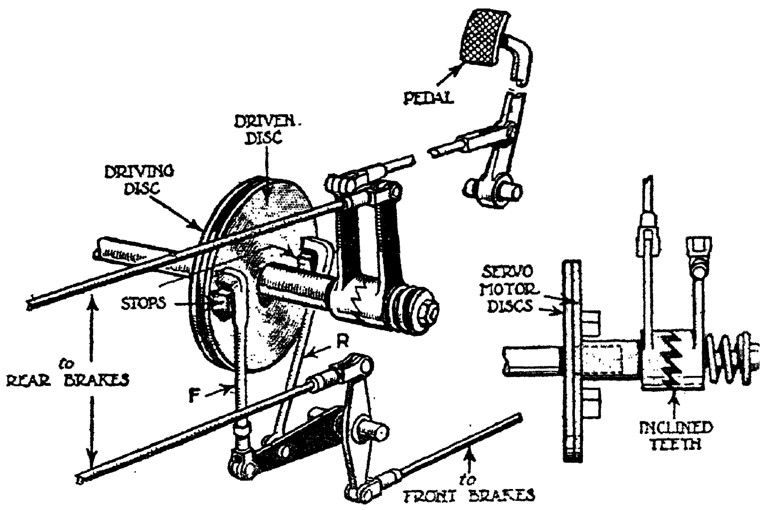


FIG. 260.- -Principle › f Servo Brakes that have been used on Rolls Royce Cars.

In the case of the Rolls-Royce brakes, there are two separate sets of brake shoes on the rear wheel drums, one set being operated by the hand lever, and another set on the front wheels, making six separate brake sets in all. The foot brake pedal operates the brakes on all four wheels, but it is so arranged that the first part of the movement of the pedal applies the brakes to the rear wheels only, but any further movement brings into

operation the Servo motor which applies all four brakes, progressively and smoothly. Fig. 260 illustrates the principle of the Servo motor.

Referring to the left-hand diagram, it will be noticed that there are two discs, the driving one being driven off a shaft in the gearbox *via* the bevel and propeller shaft, the driven one being connected by means of the two stops shown and the links *F* and *R*, to the brake-operating lever shown. The first movement of the brake pedal merely moves the upper tie-rod connected to the rear brakes. Further movement of the pedal causes the ratchet teeth between the two brake levers to separate a little.

The action of the ratchet teeth in forcing the right-hand brake disc against the driving friction disc will be clear from the right-hand diagram. Directly the two discs are in engagement, the two stops on the right-hand disc rotate, and, as shown in the lower left-hand diagram the rod *F* is pulled up, thus rotating the tee-shaped brake-operating lever below, bringing both front and rear brakes into operation. When the brake pedal is applied fully, for the car in reverse, the right-hand rod *R* is pulled up, due to the reversed direction of rotation of the friction discs. The braking effort, it will be seen, only occurs whilst the brake pedal is depressed, and its strength or amount depends upon the distance it is depressed. The greater this distance the less slipping between the discs, and the more powerful the pull on the brake rods. The driver is thus able, with little physical effort, to apply the brakes effectively. It is an important feature of the system described that it applies a greater braking effort on the rear wheels than upon the front ones; this prevents the front wheels from locking under any circumstances, and thus obviates danger.

Fig. 261 illustrates the Hispano Suiza car Servo motor, in principle; this was one of the first practical motor-car systems to be used. In this case the engine drives the worm *E*, which in turn rotates the worm-wheel *F*, from which is positively driven the drum *A*; the latter, therefore, always rotates when the engine is running. The foot brake pedal *M* is attached through a hollow shaft to a cam-shaped piece *D*, inside an expanding shoe, or segmental brake of the usual type, lying within the

driving drum *A*. When the brake pedal is depressed, the cam piece *D* expands the brake segments *B*, thus causing the drum *A* to grip them, and to rotate *B*; the latter in turn rotates the internal shaft to which it is attached, through the disc and pins shown, and rotates

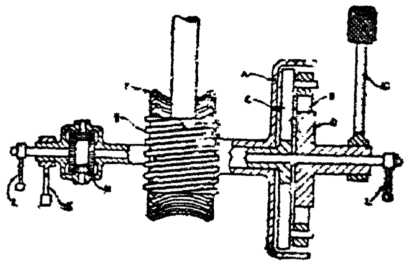


FIG. 261.—Principle ot the Hispano Suiza Servo Brake Control.

the brake levers *L* at the two ends of the shaft, thus applying the brakes. A differential compensating arrangement is provided at *H*, for the purpose of equalising the efforts on the two sets of brakes, should there be any difference of adjustments in brake mechanism; it acts in the same manner as the back axle differential gear.

Disc Brake with Servo Effect.—The Girling disc pattern brake illustrated in Figs. 262 and 263, utilizes a friction disc having a lining on each side to give the braking effect and at the same time to develop a Servo action.

Referring to Fig. 262, the road wheel has a drum-shaped pressing *1* fixed to it. This pressing has a slotted edge engaging with castellations formed on the outside of the fabric-lined disc *2*. The latter is thus rotated by the wheel drum but at the same time is free to slide sideways. A stationary carrier plate *3* is secured to the axle housing and its inner face is attached to the carrier drum *4*; the latter has its edge screw-threaded and also slotted. The slots engage with castellations on the inside of the abutment plate *5*, the thread being provided with a keeper ring *6* for brake adjustment purposes; this is effected by means of a pinion key inserted through a hole in the wheel drum. A floating pressure plate *7* is arranged concentrically with

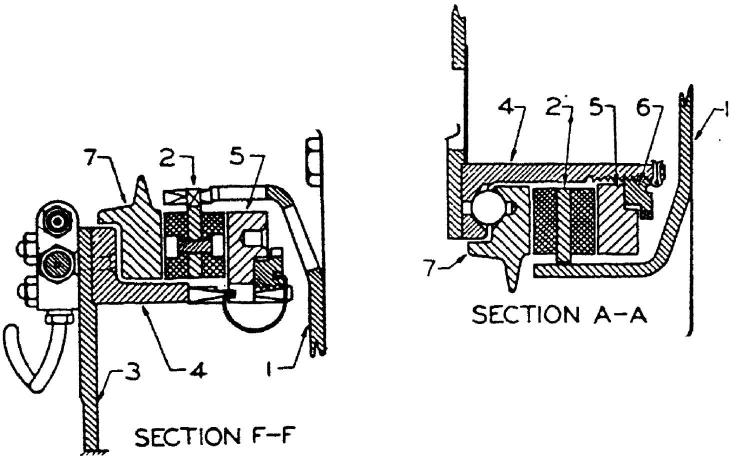


FIG. 262.—Girling Disc Type of Servo Brake.

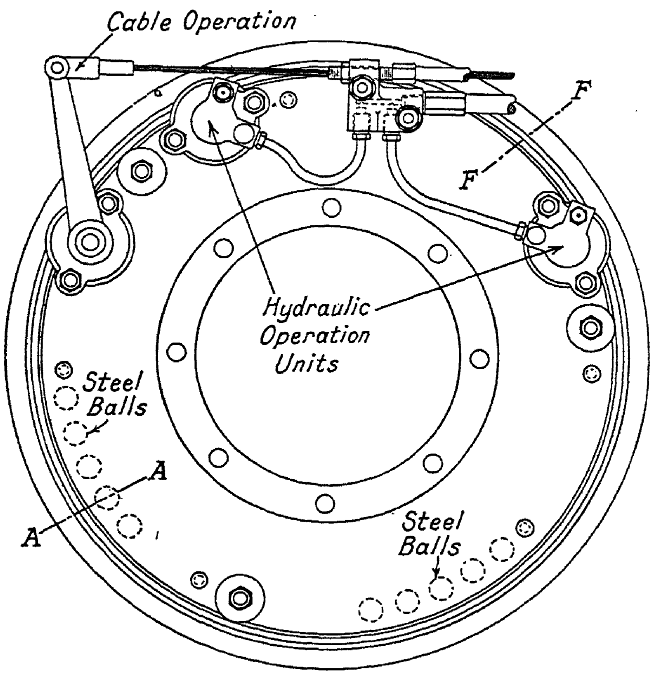


FIG. 263.—Girling Disc Brake, Showing Hydraulic Units, etc.

the carrier drum and lateral pressure is applied at two places, 90° apart by means of the hydraulic units shown in Fig. 263. There is a series of five steel balls arranged diametrically opposite to each of these hydraulic units as at AA, Fig. 263, the balls having conical seatings in the pressure plate on one side of them and in the flange of the carrier drum on the other side. Return springs are fitted to act on the pressure plate. The brake is also provided with independent cable operation means.

When pedal pressure is applied through the hydraulic units to the pressure plate, and the rotating disc is thereby clamped between the pressure and abutment plates, the disc tends to carry the pressure plate with it; the effect of this is to cause the pressure plate to endeavour to ride over the balls, whereupon the latter in their conical seatings give rise to lateral thrust and develop a powerful self-energizing or Servo action of the brake, the amount of this being governed by the angle of the conical seatings of the balls; this angle is usually about 39°.

The advantages of this design of brake are (1) Good Servo action. (2) Uniformity of pressure and wear over the whole of the friction surface. (3) Greater frictional area than the normal shoe-type of brake. (4) Freedom of the brake disc from the distortional effect which occurs on brake drums of normal shoe-type brakes.

Electric Servo Brakes. A method of using the energy provided by the electric starting motor for assisting the driver's braking effort, has been devised by the Bendix Brake Company.

The starter motor shaft is extended and carries a worm, which it drives through a free-wheeling device. When the motor is used to start the engine the worm remains stationary, because of the free-wheeling device. A reversing switch is connected with the brake pedal, and when the pedal is depressed the motor is started and turns in the reverse direction. Under these conditions the starter pinion is not engaged with the flywheel ring, but the worm on the starter shaft is driven through the free-wheeling device. Within the worm wheel driven by the worm there is a spring which expands when the brake pedal is depressed and the motor turns in the reverse direction. Frictional contact with worm wheel then tends to turn the spring

around, and in so doing adds to the braking effort exerted by the driver directly. By mounting the worm so it can slide along its shaft and away from the jaw clutch by which it is normally driven, it is made possible to apply the brake by foot in the usual way when the electric system fails. Application of the brake then causes the coiled spring to expand and to engage the worm wheel, and the reaction between the wheel and the worm draws the latter away from its clutch.

The Warner electric brake system now used in the U.S.A. employs electro-magnets in the brake drums to apply the braking effort to the friction members. It uses a small amount of current, namely, about the same as one of the vehicles tail lamps and derives its current from the battery. As the only connections between the driver's controlling lever and the drums are electric cables a very simple layout of the braking system can be employed.

Brake Compensation.—It is usual in modern braking systems to provide devices for ensuring that the actual braking effort on each brake shall be the same, irrespective of any difference of play, or adjustment, in the two sets

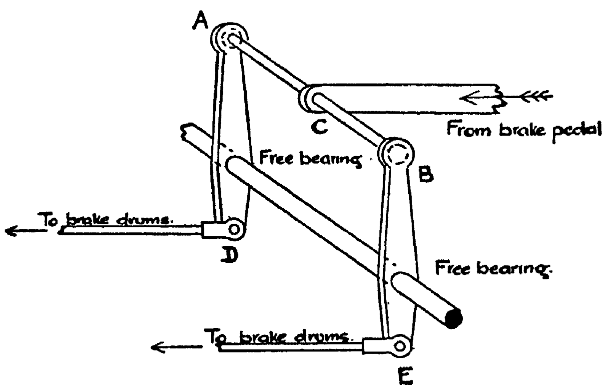


FIG. 264.—Method of Ensuring Equal Brake Pulls.

of brakes (in the case of rear wheel brakes). The simplest method of compensation is by means of the double lever system shown in Fig. 264. Here the braking pull due to the pedal is applied to the centre *C* of a rod *AB*, whence it divides between the two levers *AD* and *BE*, connected,

respectively, to the two rear brakes. If there is more play, in say the left-hand brake system, then its operating

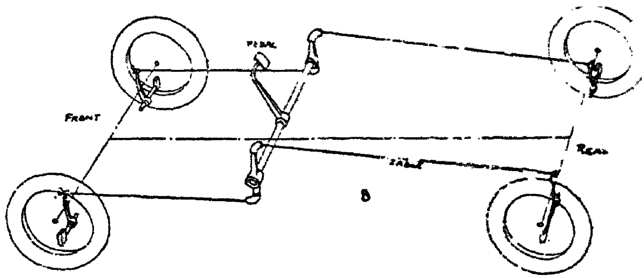


FIG. 265.—Simple Four-Wheel Braking Arrangement.

lever *AD* will swing forwards under the pedal effort until all of this play has been taken up, when both ends *A* and *B* will be pulled equally, thus ensuring equal braking

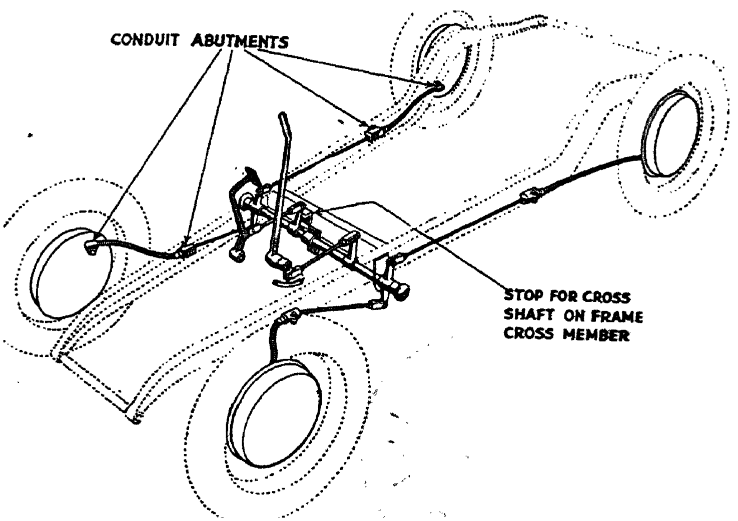


FIG. 266.—Bendix Braking System.

efforts. Sometimes, as we have seen, a differential gear is employed to compensate the braking efforts. The same methods are employed both for front wheel brake operation and also for the actuation of all four wheel brakes.

Fig. 265 illustrates a typical layout of a simple braking system. There is no special compensation means employed for both front and rear wheels, but each brake is adjustable, individually. Fig. 266 shows the Bendix braking system in which flexible cables are used to operate the brakes. The hand brake, it will be observed acts upon all four wheels, but is not affected by the foot brake movement.

Brake Drums.—In the past the majority of brake drums have been made in the form of steel pressings. It has been found, however, that this type of drum not only tends to become scored but also gives rise to brake 'squeaks' owing to the vibrations of the metal; moreover the cooling of the drum is not very effective—for there is

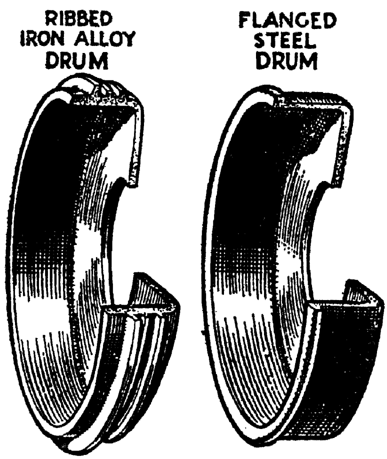


FIG. 267.
Ordinary Type of Brake Drum (on right) and Improved Type (on left).

usually a considerable amount of heat generated by the friction lining-action on the inside of the drum.

A later improvement in brake drums is to make them in the form of a nickel-iron casting with suitable air-cooling ribs as shown in Fig. 267. Aluminium and also magnesium alloy cast ribbed drums, fitted with hardened or nitrided steel liners are used on some of the larger cars and also on racing cars.

Hill Holding Devices. When a car is to be started from rest on an incline it is necessary to release

the hand brake simultaneously with the engagement of the clutch and acceleration of the engine. This is not always an easy procedure, more particularly with drivers of limited experience, so that devices have been produced for releasing the brake automatically at the moment the clutch is engaged. It is only possible to describe one or two of these 'hill-holding devices' or automatic sprags,

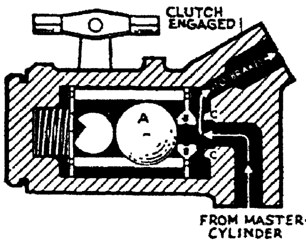


FIG. 268.

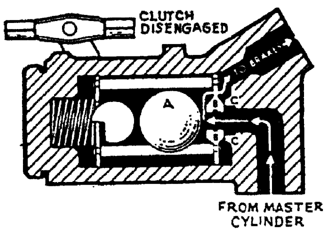


FIG. 269.

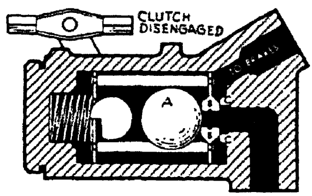


FIG. 270.

namely the 'No Rol' one shown in Figs. 268, 269 and 270, and the Lockheed device in Fig. 271.

The former device consists of a special gravity-controlled valve in the hydraulic brake system, automatically coming into action on an up-grade. There is also another valve in the system which is worked by a lever arrangement connected to the clutch pedal mechanism.

When stopping on any incline, no matter how steep, the driver simply applies the brake and disengages (or throws out) the clutch in the usual manner. When the car comes to a complete stop, an automatic check device retains the hydraulic pressure in the braking system which was developed when the brakes were applied. This hydraulic pressure holds the brakes applied as long as the clutch pedal is held depressed, enabling the driver to take his

foot off the brake pedal and use the right foot for acceleration purposes. It is therefore unnecessary for the driver to keep his right foot on the brake pedal whilst the car is stopped. When ready to proceed up the hill the driver depresses his accelerator and engages the clutch when the brakes are automatically released and the car moves forward, so that there is nothing new for the driver to learn.

Referring to the illustrations, these show a cross-section of the ' No Rol ' unit and depict the valve body and ball case containing the ball and cam shaft.

(Fig. 268) When the clutch is engaged, valve *B* is held away from seat *C* permitting free passage of brake fluid between brakes and master cylinder regardless of position of ball *A*.

(Fig. 269) This illustration shows the action that takes place in valve when clutch is disengaged (thrown out). . . . When clutch is disengaged, valve *B* is pressed against seat *C*. Brakes may be applied or released while car is in forward motion. The NoRol is not effective while car is moving forward.

(Fig. 270) With clutch disengaged and brakes applied on an up-hill stop, gravity will cause ball *A* to seat against valve *B*. Brake pressure is held within the system after removal of foot from brake pedal. Engagement of the clutch releases brake pressure as the car is started.

In the case of hydraulic braking systems the car can also be held on steep inclines or gradients by means of a device produced by the Monarch Governor Company, Detroit, Michigan, U.S.A.

It consists essentially of a valve which is installed in the hydraulic line of the braking system a short distance from the master cylinder, ahead of the junction of the lines leading to the front and rear wheel cylinders. When this valve is closed the brake fluid is trapped in the wheel cylinders and the car cannot move. Release of the valve is effected by merely flicking the finger lever on the steering post, by means of a special auxiliary valve.

The Lockheed automatic hill holder shown in Fig. 271 is another alternative device for use with hydraulic braking systems. It is comparatively simple in design and embodies a gravity-operated ball guided in a cage and a valve interconnected with the clutch pedal. The arrangement is such that the ball valve closes whenever the car is on an upward

grade and so long as the clutch pedal is depressed the fluid is prevented from returning to the master cylinder. Pressure is thus maintained and holds the brakes on, but directly

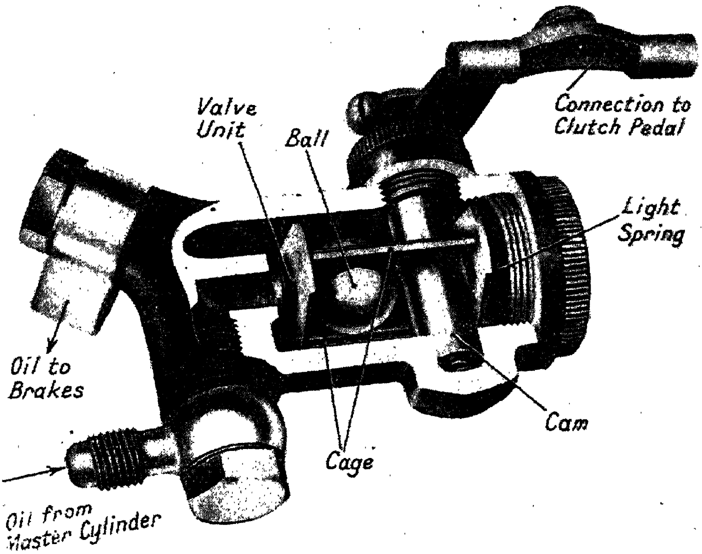


FIG. 271.—The Lockheed Hill Holding Device.

the clutch is engaged the valve is lifted off its seating so that the brakes are released as the clutch is engaged, thus dispensing with the need for using the hand brake when restarting on a gradient.

The Care of Brakes.—Although modern brakes require little attention as a rule, it is important that they receive this regularly. The amount of movement of the foot pedal and hand brake lever before the brakes come on is a good indication of the necessity, or otherwise, of adjustment. The only adjustment necessary in the case of earlier cars is an occasional turn of the turnbuckles or hand screws on the longitudinal rods between the brake pedal or hand lever and the brake drums. *It is quite easy to over-adjust the brakes, however,* so that they actually bind. A binding brake causes loss of power. It can readily be detected by feeling the brake drum soon after stopping.

If hot, then the brake is binding. The best plan is to jack up each wheel in turn—or, better still, all wheels—and adjust each brake separately, until it is just free when the wheel is spun, and the brake pedal or lever disengaged. Most mechanical brake mechanisms have a number of pin joints and rocking arms; these should be kept well greased, and free from mud or dirt; in this respect the enclosed internal expansion brakes are an advantage over the earlier external contracting type.

If, when the brake shoe, rod or cable adjustment is fully taken up, the brakes are not satisfactory in action, it is a sign either that the cams or shoe ends in the drums are worn, or that the shoes require re-lining. Sometimes, by soldering, screwing or pinning steel liners to the brake shoe ends, on which the cam acts, the former trouble may be overcome. The process of re-lining a brake is quite simple, the new lining being riveted in position with soft copper or aluminium rivets.

The Engine as a Brake.—The engine, when switched off, may be used as a brake when descending long or steep hills. The maximum braking effect is obtained in low gear with the throttle wide open, preferably with an extra air valve also wide open; this prevents fuel wastage. In this case the rear wheels and transmission have to turn the engine against the pumping resistance of the pistons. It is inadvisable to make a habit of using the engine as a brake on all occasions, particularly when the throttle is nearly closed, as the engine oil is apt to be drawn past the pistons due to the suction created, and thus to cause more rapid carbonising. Moreover, explosions in the silencer may occur when switching on.

Notes on Four Wheel Brake Design.—When brakes are fitted to the front wheels, it is necessary to consider the design of the front axle, steering and springs at the same time. The additional stresses caused by applying powerful braking torques or forces, to the front wheels must be allowed for in the members mentioned. It is necessary to stiffen up the front axle portions between the steering pivots and spring pads; this is usually carried out by making these parts of tapering circular section, instead of the usual I beam one. The application of front wheel

brakes causes a powerful torque, which tends to bend the front ends of quarter elliptic springs upwards, and their anchorages downwards, *i.e.*, increases the deflection. With semi-elliptic springs the front portion tends to bend upwards, and the rear portion downwards, the resulting tendency being that of a deformation. In order that the front brake application shall not affect the steering, the operating cam must lie on the steering pin axis, as previously explained ; further, it becomes necessary to use *centre point steering* in order to minimise the ill effects of wear and bad adjustment of the brakes. Without centre point steering, the effect of the application of the brakes, when the wheels are being steered, is to cause a serious drag on the steering. In some cases, it is arranged for the outside front wheel brake only to be applied when the car is being steered around a curve, in order that the two wheels should not be braked equally, thus causing skidding of one, or both.

Another important point in the operation of four wheel brakes is that the rear braking effort should always be greater than that of the front wheels. If the latter were to experience a greater braking effort, or to lock first, there would be a serious danger of skidding. The rear foot-operated brakes are, therefore, either applied first, or are given a greater braking leverage than the front brakes, so as always to be the more powerful in action.

When the car is in reverse gear, the brakes should operate equally well, as when the car is held on a steep hill. With Servo brakes it is, therefore, better practice to work the Servo motor, or friction disc, or drum, off the transmission, *i.e.*, off the rear wheel indirectly, as in the Rolls-Royce system described, rather than from the engine direct. The braking effort is then proportional to the speed, not of the engine, but of the car—which is as it should be.

Brake Efficiency Indicators. The rate of bringing a car to rest, from a given level road speed, is a measure of the braking efficiency of the car's brakes. Expressed more precisely, the rate of deceleration—which is the opposite to acceleration—gives an accurate indication of the brake efficiency.

Instead of timing the operation of stopping a car from

a given speed and thus obtaining data for estimating the rate of deceleration, special devices, known as *decelerometers* are generally employed for this purpose. These usually depend upon the principle of a vertical pendulum which deflects from its vertical position when its mounting is accelerated or decelerated; the angular deflection provides a measure of the amount or rate of acceleration or deceleration.

A typical decelerometer or brake efficiency indicator is the Ferodo one shown in Fig. 272, which uses a metal ball which is free to roll in a tube instead of a pendulum.

The force with which the ball tends to roll forward is proportional to the deceleration, and if this force is balanced, by setting the tube at an angle up which the ball just rolls, this angle becomes a measurement of the particular rate of deceleration.

The method by which advantage is taken of this principle is shown diagrammatically in Fig. 272.

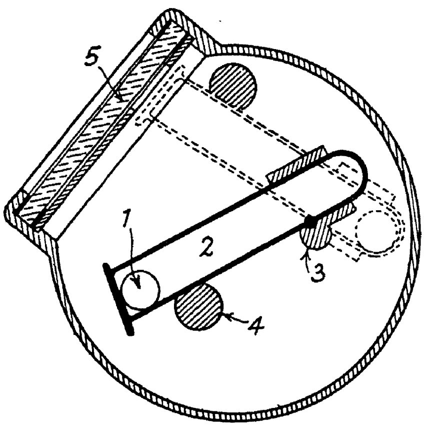


FIG. 272.
Showing Principle of the Ferodo Brake Efficiency Tester.

The ball shown at '1' is enclosed in the tilted tube '2.' The tube itself is supported on a pivot at '3,' and is retained at the angle shown by resting on the stop '4,' against which it is held by the weight of the ball. When the deceleration of the vehicle is sufficient to cause the ball to roll up the tube and past the pivot, the ball's weight causes the whole tube to tilt over into the position in which it is shown dotted. The end of the tube carries

a coloured indicating signal, which signal can be seen through the window '5.' It will be seen, therefore, that on the signal being given the driver knows that his brake has reached the particular degree of efficiency predetermined by the angle at which the tube is set. In the standard indicator three such combinations of pivoted tubes are fitted at different angles, so that three distinct signals of brake effectiveness are given. The instrument is designed for stopping distances from 20 m.p.h., the three windows giving stopping distances of 22, 27 and 45

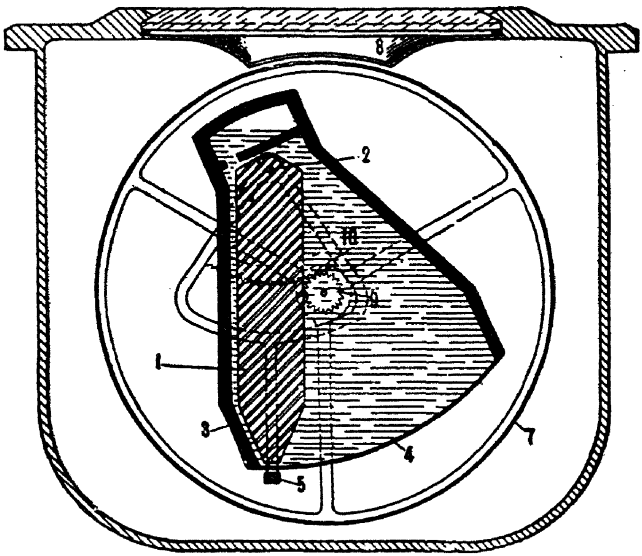


FIG. 273.—The Ferodo-Tapley Brake Testing Meter.

feet, corresponding to braking efficiencies of 60, 50 and 30 per cent., respectively. The zero or level setting of the instrument is carried out by means of a pointer, seen through a separate window, this pointer being operated by a pendulum not shown.

It should be observed that the signals remain in their respective windows until the tubes are re-set after a test, by means of a small lever on the side of the instrument.

Another instrument, known as the Ferodo-Tapley Brake Testing Meter, Fig. 273, gives continuous readings of brake

efficiency and also the distances in which the vehicle can be stopped from a speed of 20 m.p.h. This instrument is a decelerometer operating upon the pendulum principle and provided with a method of damping out road vibration and similar effects which, otherwise, would prevent steady and accurate readings being obtained.

Referring to Fig. 273, the pendulum, which is a powerful permanent magnet, is enclosed in the box (3) the latter being filled with a damping fluid and sealed. The pendulum can only move in the fluid and is thus well damped. A thin curved plate (4) forms the bottom of the box. Outside the latter, on pivots mounted in line with those of the magnet pendulum, is a light armature (5) so arranged that by magnetic attraction from the pendulum it will indicate the exact position to which the pendulum moves. By means of a toothed segment and pinion, at (10) and (9), respectively, the movement of the armature is magnified and transferred to the revolving indicating scale (7). The latter encircles the damping box and is seen through the window (8). The instrument also embodies a mechanism for holding the scale at its highest deflection position during a test. This consists of a ratchet-toothed wheel mounted on the same spindle as the scale, having a pawl engaging with it. By means of a small lever the pawl can be lifted out of engagement allowing the scale to return to zero.

The instrument can be clamped to the instrument panel or any other convenient part of the vehicle to be tested, by means of a universal clamp mounting. In reference to the position of the instrument, as shown in Fig. 273 the direction of motion of the vehicle is to the right.

Brake Efficiency Values. In reference to the values obtained for brake efficiencies either by direct measurement or decelerometer readings, if the value is below 30 per cent. the brakes are in poor condition ; 40 per cent. corresponds to two-wheel brakes in good condition ; 45 per cent. to the upper limit achieved by the best two-wheel brakes ; 50 per cent. to four wheel brakes in good condition ; 60 per cent. to four wheel brakes in very good condition ; 70 per cent. to these brakes in excellent condition ; 80 per cent. to the safest degree of efficiency at which four-wheel brakes can be maintained. Above 80

per cent. on dry roads there is a risk of wheel lock and skidding.

Brake Testing Apparatus.—The ordinary methods of adjusting the brakes where four-wheel brakes are used depend, to a large extent, upon the skill of the individual. It frequently happens that these adjustments are not carried out very accurately, so that when the brake-pedal is depressed the wheels are braked unevenly.

The most satisfactory method of adjusting brakes is to employ brake-testing plant, such as those now marketed by Messrs. Harvey Frost, G. E. Equipment, Smith and Bendix Perrot. The apparatus supplied by these firms although differing in actual design, depends upon the same *principle*, viz., that of measuring the torque, or pull, at the periphery of the tyre, required to just rotate the wheel

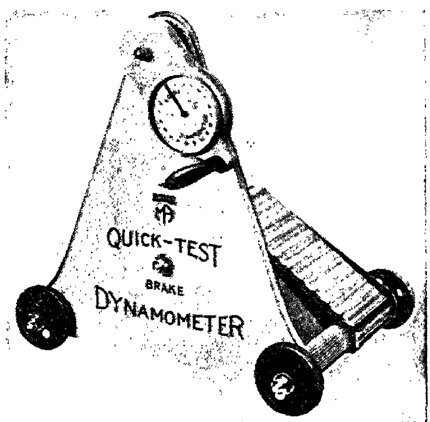


FIG. 274.—Smith Quick-Test Brake Dynamometer.

when the foot-brake pedal is depressed; a special device is supplied for holding the foot-brake pedal in the position of normal application.

The Smith brake testing machine, illustrated in Fig. 274, has a kind of caterpillar track, giving an effective tyre grip, driven through suitable gearing by means of a handle which contains the measuring dial. The resistance to the rotation of the wheel which is produced by the application of the

brake is transmitted to the dynamometer handle through the gearing to the caterpillar track, and is measured by the handle effort. The wheel must be jacked up, the brake pedal held depressed by means of a strut and the brake tester placed behind the wheel under test. Each size of car has its own corresponding dial reading, ranging from 3 to 4 for an 8 h.p. car to 7 to 8 for a 20 h.p. one, for efficient braking conditions.

The Harvey Frost apparatus operates in a similar manner; it has a dial type of gauge for giving the brake effort readings.

An example of a more elaborate brake-testing plant is shown diagrammatically, in Fig. 275. In this case power for turning each of the road wheels against the action of the brakes is derived from a separate electric motor.

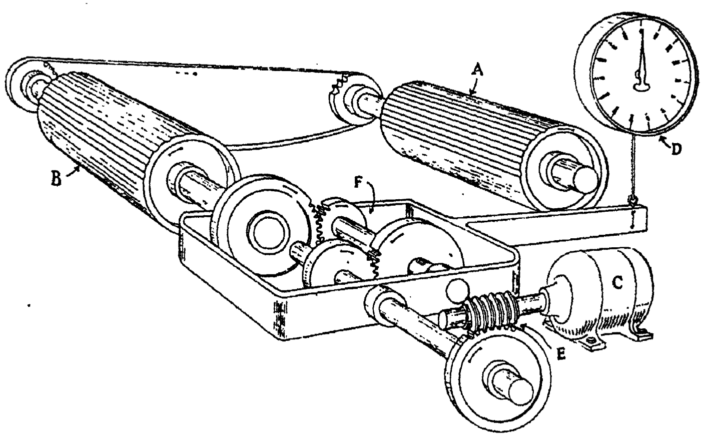


FIG. 275.—Bendix Perrot Brake Testing Machine.

The turning effort is measured by means of a torque arm, similar to that used for measuring the torque of petrol engines in the case of power brakes.

Referring to Fig. 275 each wheel of the vehicle rests upon a pair of grooved rollers *A* and *B*, the grooves representing the average road resistance to the tyres. The wheel *B* is driven through the gearing *E* and *F* from an electric motor *C*. The two rollers *A* and *B* are connected together by a chain which passes over sprockets on the rollers. When the brake on the wheel is applied by

means of a brake-depressing strut, and the electric motor is switched on the gears will be rotated so as to tend to turn the rollers A and B. The effect will be to cause an opposing turning movement on the hinged frame carrying the gears and motor causing the right hand side to move downwards. The pull at the end of the right hand side is then measured on the spring balance dial D. This pull, multiplied by the radius of the frame torque arm, gives the braking torque.

Each wheel of the vehicle is tested separately and in order to prevent the whole vehicle from moving bodily, one end is anchored by means of a chain to a post fixed in the ground. In this connection the greater the pressure on the brake pedal the more will the vehicle tend to climb the rollers.

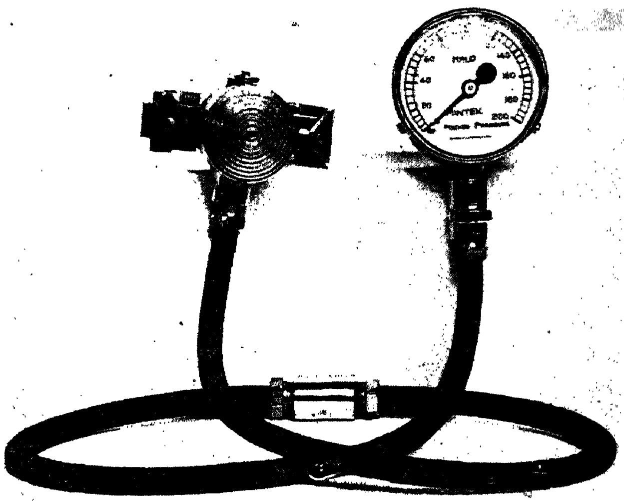


FIG. 276.—The 'Pressometer' Brake Pedal Pressure Testing Device.

The four brake testing units are generally arranged in pits, with the rollers just above the ground level, so that vehicles can drive straight on to the rollers.

When testing the braking efficiency of a motor vehicle it is important to know what is the actual value of the force exerted upon the brake pedal, since under ordinary

operating conditions the force applied to the latter may vary considerably, depending on the physical conditions as well as the weight of the driver.

A suitable instrument for measuring the force exerted on the brake pedal is that known as the 'Pressometer' (Fig. 276).

In basic principle it consists of a very small cylinder, $2\frac{3}{8}$-in. long and $1\frac{1}{2}$-in. diameter overall, fixed on the foot pedal of the motor vehicle, in such a manner that the pressure exerted when applying the brake depresses the piston. The pressure is transmitted by oil in the circuit, through a port in the cylinder wall and two standard 'Lockheed' hose connections, to a small dial, with a pointer, which is carried in any desired position on the windscreen, attached by rubber suction grips. The dial is graduated in degrees of 5 lbs. each within the range of 0–200 lbs. total pressure exerted, and the exact figure can be read at a glance. A priming valve is also provided at the base of the piston to make good any slight leakage.

CHAPTER IX

THE ROAD WHEELS

The rear road wheels are called upon to transmit the driving torque of the transmission, to support the weight of the rear end of the car, and also to withstand severe road shocks, both vertically and sideways. In addition, they must be provided with hubs for housing the bearings, and with suitable rims to hold the tyres. Modern road wheels are invariably of the readily detachable pattern; the detachable rim type is still found on certain cars on the road; the latter has the advantage of lightness, since only a spare rim, and not a complete wheel, need be carried.

It is usual to mount the necessary ball or roller bearings in a separate hub, of the totally enclosed lubricant-filled type; this hub is not detached from the axle, when the wheel, or rim, is removed for tyre changing purposes.

There are four principal types of road wheel which have been used, although to-day only two or three types are in existence. The former include (1) The Artillery Type. (2) The Hollow Steel Spoked Type. (3) The Pressed Steel Disc Type. (4) The Wire Spoked Type.

The Wooden Wheel.—This was one of the earliest types and at one time widely used upon American, and some Continental cars. In this case there is usually a wooden or steel tyre rim, connected to the steel hub casing flanges by means of ash, or more commonly, hickory, spokes, in a somewhat similar manner to ordinary cart wheel practice. The wooden wheel is usually arranged with a steel outer rim on which the detachable tyre rim is mounted, and fixed by means of five or six bolts and nuts. It possesses the advantage of strength combined with lightness, and is of pleasing appearance. It is not as strong, laterally, as the other types, and the wheel is, therefore, more likely to be forced out of truth by a

side impact, as when it runs slantwise into a kerb. It had the disadvantage of shrinkage effects when used for long periods under tropical heat conditions.

The Hollow Steel Spoked Wheel.—This was at one time, one of the most popular types in this country. It resembled the wooden type in appearance, but was made entirely of thin gauge steel, by welding together steel pressings. Usually, ten hollow steel spokes were provided.

In the Sankey wheel construction, Fig. 277 (1) and (2)

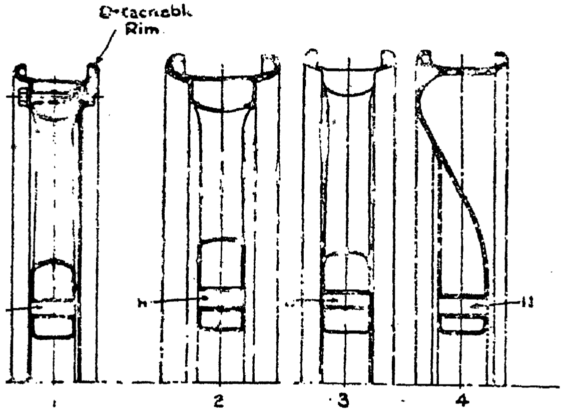


FIG. 277.—Various Types of Steel Wheels.

1. The Sankey Hollow Spoked Wheel, with Detachable Rim.
2. The Sankey Pressed Steel Hollow Spoked Wheel.
3. The Goodyear Pressed Steel Hollow Spoked Wheel.
4. The Sankey Pressed Steel Disc Type of Wheel.

Note.—H denotes bolt-holes for attaching to hub.

the hub spokes, and wheel rims are made in two pressings, or hollow half-sections of the wheel—similar to the sections obtained by cutting a complete wheel through a central plane perpendicular to the axis, but parallel to the plane of the wheel. These two half sections are welded together so as to form one complete wheel. The other part, or unit, used forms lateral webs in the spokes; it is inserted between the spokes before they are welded together.

These wheels were made in the ordinary fixed rim and also the Warland detachable rim type; in the former

case, the wheel is attached to the wheel hub (on the front or rear axles) by means of six bolts and nuts passing through holes in the wheel's central portion.

The Goodyear steel spoked wheel is made in three parts, namely, a complete, continuous tyre rim, welded between two half-section steel pressings, forming the spokes, felloe and nave, as shown in Fig. 277 (3); this form of construction gives a stronger rim, the lateral strength of which is greater than in the welded type. In both of the examples illustrated, the hub securing bolts pass through stiffening tubes connecting the two wheel faces at the centre; the act of tightening up the wheel-securing nuts does not, therefore, distort the two wheel sides.

The Steel Disc Wheel.—Although the steel disc wheels, unless of heavy gauge, are not so strong, laterally, as the spoked types, they are cheaper to make, and easier to keep clean; in many cases, also, they enhance a car's appearance.

Fig. 277 (4) illustrates the construction of the Sankey disc wheel, which has been used on small cars, and shows how

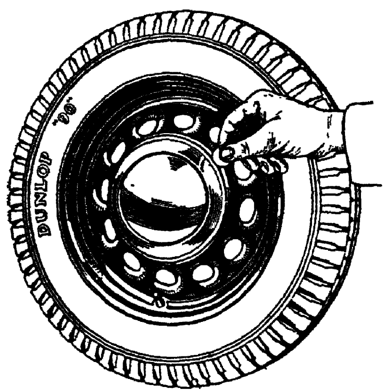


FIG. 278.—Pressed Steel Large Hub Wheel, showing method of removing Hub Cover.

the separate tyre rim is welded to the dished pressing which takes the place of the spokes, and which forms one side of the hub. There is also an annular disc of steel (shown on the left in Fig. 277 (4)), which forms the other side of the hub, and at the same time acts as a lateral and vertical stiffener of the other disc. These wheels are rather heavier than the hollow steel and wooden spoked types; their use has now been confined to certain commercial

vehicles. The disc wheel is apt to cause 'drumming' and to accentuate the noises made by the transmission.

The pressed steel wheel which has largely replaced the earlier steel spoked and plain disc wheels, has the advantage of comparative lightness and rigidity, combined with a pleasing appearance, for the perforations (Fig. 278)

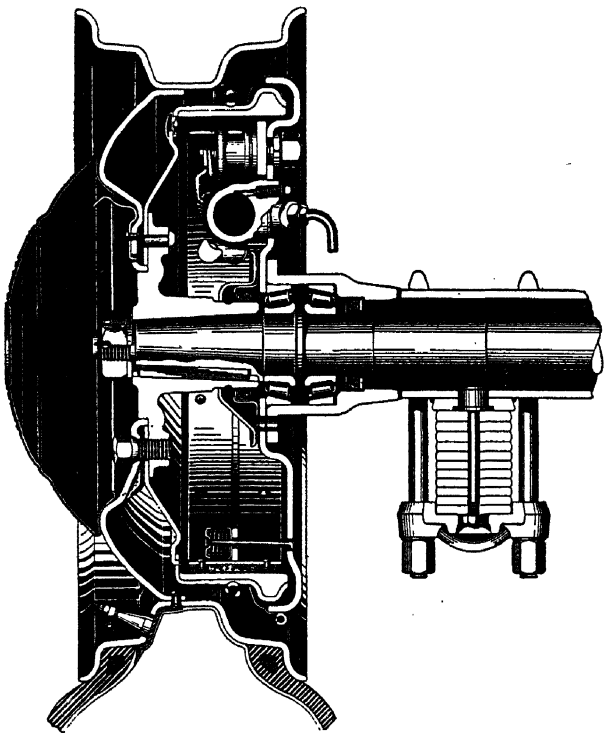


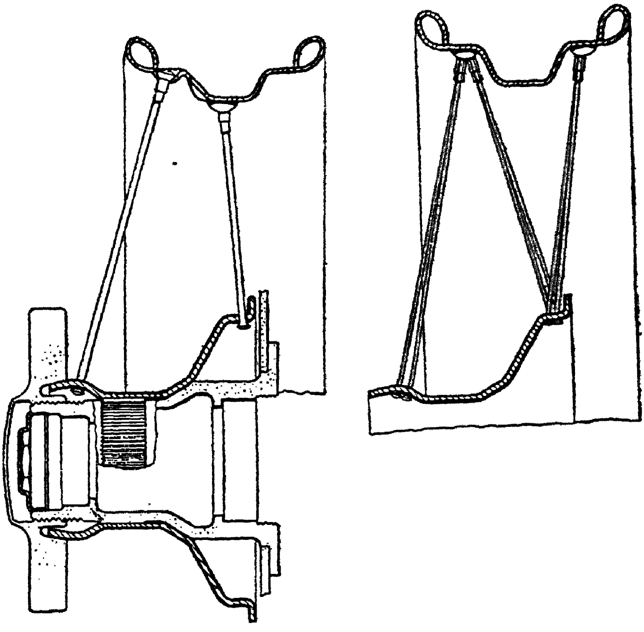
FIG. 279.—Pressed Steel Rear Wheel for Large Car.

give the wheel a lighter effect than the plain disc pattern. This design of wheel has a large Magna type hub with the wheel security bolts and nuts housed inside ; a chromium-plated cap readily removable with a coin or screwdriver acts as a dust cover and, incidentally, adds to the neat appearance of the whole wheel. This type of detachable wheel fits in with the use of large-section low pressure tyres and is readily cleaned.

An example of a well-base rim, built-up (welded) steel

wheel is given in Fig. 279 which illustrates the Packard rear wheel and semi-floating rear axle. The wheel is readily detachable, being held to the coned axle member by a number of studs and nuts—one of which is shown below the axle nut. A large chromium-plated cover encloses the wheel nuts.

The Wire Spoked Wheel.—This type, which has been much used from the earliest times on motor-cars and motor-cycles, is undoubtedly the strongest one for its weight, of any. In this case the tyre rim is attached to the wheel hub, by means of numerous high tensile steel spokes, arranged in two or three sets or rows



(A) (B)

FIG. 280.—Types of Wire Spoked Wheels.

The spokes, which are the only connection between the wheel rim and hub have to perform three distinct functions as follows: (1) To take the vertical load or weight of the car; (2) To withstand the driving torque; and (3) To withstand the braking torque. Although for the former

purpose the spokes should be arranged radially, yet, in order to take the forces due to (2) and (3) they must be placed tangentically. A compromise is therefore adopted, in practice, by arranging the spokes in pairs each member of which slopes alternatively in the opposite direction, tangential to the hub. This arrangement applies to each row of spokes, at the front and back of the hub; for three rows of spokes a similar plan is adopted. Fig. 280(A) shows a typical two-row wire wheel* and hub, and Fig. 280(B), a three-row wire spoked one. It will be observed that the spokes are inclined, so as to form with the hub a rigid triangle for resisting lateral loading effects. The

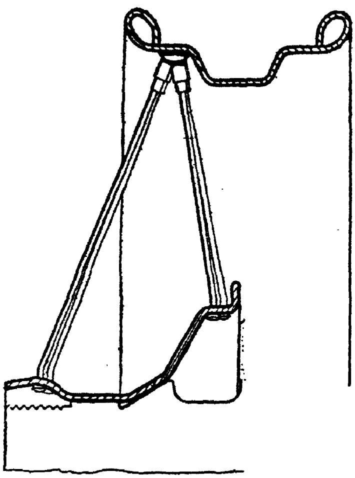


FIG. 281.—Offset Rim of Wire Wheel to give Centre Point Steering Action.

wheel is detachable and for this purpose its hub is splined inside, so as to fit over the non-detachable splined hub on the wheel axle. In order to locate the wheel it is provided with a conical section in its hub which bears upon a similar outer cone on the non-detachable hub. The wheel is held in place by means of a combined knock-on nut and cover, having two lugs or arms. In order to prevent the cap nut from loosening and unscrewing, the right-hand wheel nut is given a right-handed thread and the left-hand one a left-handed thread; the forward direction of rotation of the wheel therefore tends to screw the cap-nuts up.

* The *Autocar*.

In order to bring the axis of the king-pin into the same plane as the centre plane of the tyre so as to obtain centre point steering action it is usual to attach the rim ends of the spokes towards the outside of the wheel rim, i.e.,

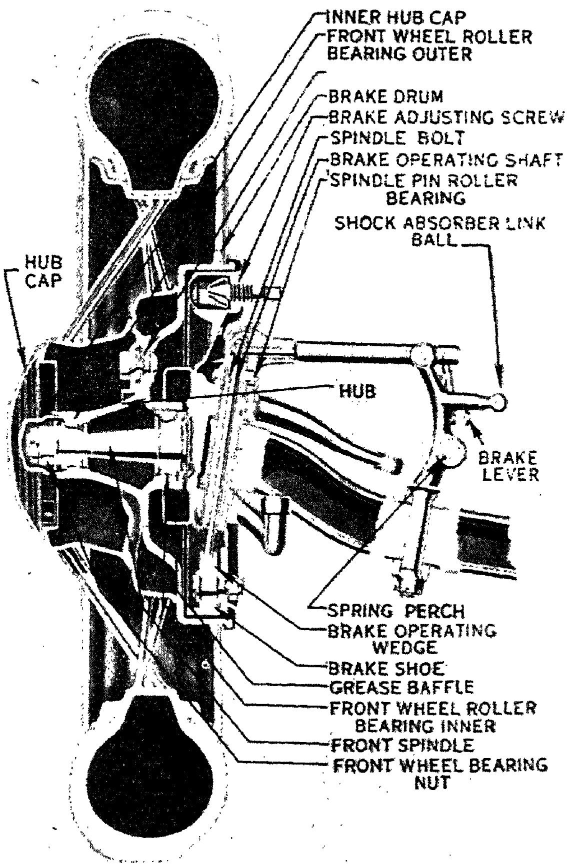


FIG. 282.—The Ford Wire Spoked Front Wheel, showing Hub Construction.

between the centre of the rim and its outside edge. In some cases, notably in the wheels of certain high speed sports cars, the two sets of spokes from either side of the wheel hub meet the wheel rim in a single circle towards the outside of the rim, as shown in Fig. 281, thus giving

a good deal of overhang of the rim on the inside of the wheel. In the well-known Rudge-Whitworth wheel, there are three rows of spokes, which are held to the central hub by means of their enlarged heads, and are secured to the tyre rim by screwed nuts, or nipples ; the latter are also used to tension the spokes equally. The hollow metal hub is provided with a series of internal teeth or serrations, which fit on corresponding serrations on the axle hub ; these serve the purpose of locking the wheel to the axle hub, a large nut being provided for tightening the wheel hub on to the latter ; the serrations, it will be seen, act as keys, to lock the wheel to the axle hub. The wheel can be quickly detached by hammering, on the projections provided, the lock nut in order to start it, in the normal forward direction of rotation of the wheel, and then unscrewing it as far as it will go ; the wheel can then be pulled off, if the axle has been jacked up. The serrations must be kept clear of dirt, and well greased ; the groove in the locknut should be filled with oil before replacing, and the locknut hammered home afterwards.

The wire spoked type wheel has been much used on the larger model cars, and also on racing cars, on account of its great strength, lateral rigidity and relative lightness. It is, however, more difficult to clean, and the spokes, formerly, were apt to rust under the nipples, and to break eventually ; rust-proof spokes are now employed.

Light Car Wheels.—The wire-spoked type of wheel was also fitted to certain small cars, as it is the lightest form of wheel in present use.

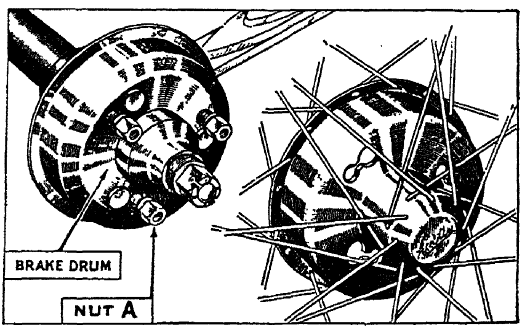


FIG. 283.—The Austin Seven Wheel.

The wheels are provided with dished-type metal hubs, flanged on one side to carry the spokes on the inner-side of the wheel.

These hubs are so shaped that they fit over similarly shaped hubs on the wheel brake drums. They are secured to the latter by three or four studs and nuts. Fig. 283 shows the hub portion of the detachable wire wheel of the earlier Austin Seven car. The wheel hub is provided with slotted holes, enlarged at one end so as to slip over cylindrical projections formed on the brake drum. These relieve the studs and nuts *A* of any torque, and render the fitting of the wheel more easy.

Heavy Wire-Spoked Wheels.—Certain models of Ford cars were equipped with all-metal wheels of the wire-

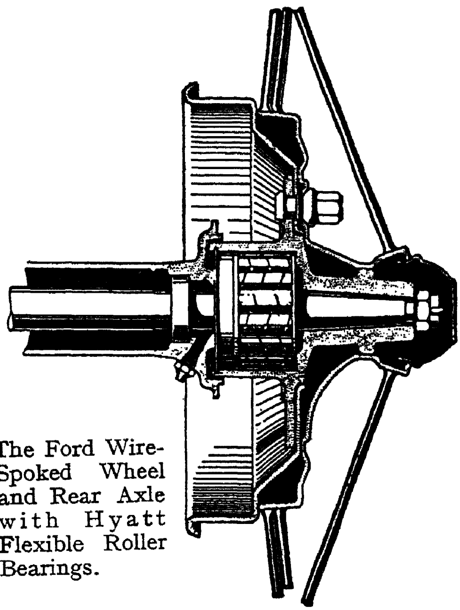


FIG. 284.—The Ford Wire-Spoked Wheel and Rear Axle with Hyatt Flexible Roller Bearings.

spoke type. Each wheel has thirty steel spokes, each of about $\frac{1}{4}$-inch diameter, arranged between the rim and the hub, as shown in Fig. 284. Owing to the increased strength of the heavy-gauge spokes used, a smaller number of the latter are required.

The wheels are detachable, being held in position on

the axle hub drums by five studs and nuts, the former passing through the inside part of the detachable hub plate. In this way the nuts are kept inside the outer row of spokes. An advantage of this type of wheel is that it is particularly strong, light in weight, and easily cleaned.

The Magna Wheel.—This is another popular type of built-up steel wheel. It is characterised by its large diameter hub, the outer part of which is usually covered with a chromium-plated cap. The hub is of hollow construction, with five holes for attachment to the studs on the axle discs (on the inner face), and five larger sector-shaped holes for access to the wheel holding nuts on the outside.

The hub of the wheel shown in Fig. 285 is joined to the wheel rim by means of steel spokes, these are shorter than the usual type of wire wheel so that there is rather

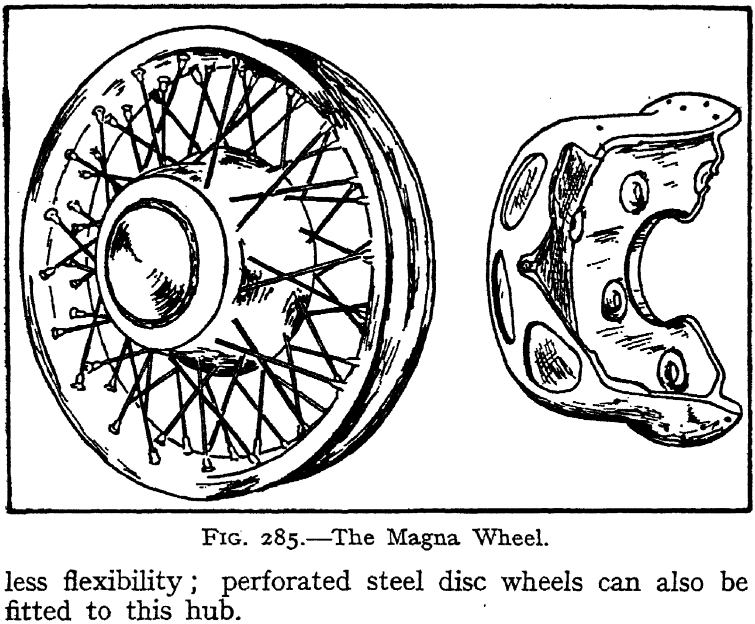


FIG. 285.—The Magna Wheel.

less flexibility; perforated steel disc wheels can also be fitted to this hub.

Elektron Wheels.—In order to save an appreciable amount of weight—in the unsprung members—the use of the extremely light magnesium alloy, known as Elektron,

has been advocated. In this connection it is of interest to note that the alloy in question is less than two-thirds the weight of aluminium, but is much stronger than this metal. One of the leading motor-bus companies has carried out tests with a number of buses fitted with Elektron wheels and after a period of service extending to over 80,000 miles of running on London roads, the wheels in question were quite satisfactory.

The weights of the front and rear wheels of a motor vehicle in steel were 154 and 264 lbs., respectively. When these were replaced by Elektron the weights became 48 and 92 lbs., respectively; these wheels, it will be seen, were about one-third the weight in steel.

Light Alloy Car Wheel.—The use of a light alloy for car wheels of good appearance is illustrated in Fig. 286 in the case of the "Airlite" wheel. This is designed with short spokes which produce an air circulation effect for cooling the brake drums. Each wheel is from 3 to 6 lbs. lighter than the usual steel pattern, so that the unsprung weight

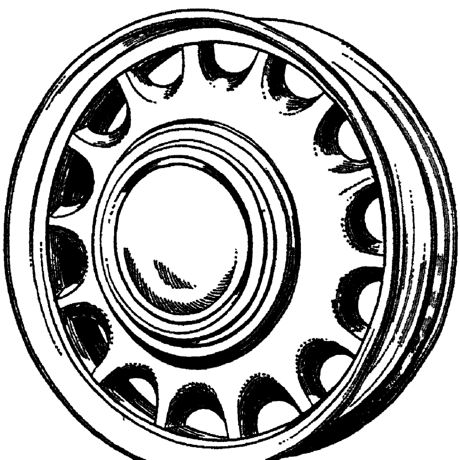


FIG. 286.—Light Alloy Wheel.

of the car is reduced proportionately. The construction of the wheel is such that it has about the same strength and rigidity as the normal pattern.

Tyres.—It is not proposed to discuss in detail the construction and fitting of motor tyres, as this subject hardly comes under the mechanical heading, but a few remarks are appended on the types of tyre in present use.

The *Fabric or Canvas* motor tyre, which was much in vogue a few years ago, was not altogether satisfactory, owing to the continuous flexion and chafing of the fabric threads; its life was usually limited to 6,000 to 9,000 miles.

The *Cord* method of construction, now in more general use, employs stronger plies of the material in successive layers, each formed of one layer over the other, with a thin rubber sheet between. Each ply consists of a number of small cords, themselves composed of several strands twisted; these cords are held in position by slender and widely spaced cross-threads. The cord tyre is, therefore, much stronger than the fabric type, and lasts much longer; its normal road life is from 10,000 to 15,000 miles.

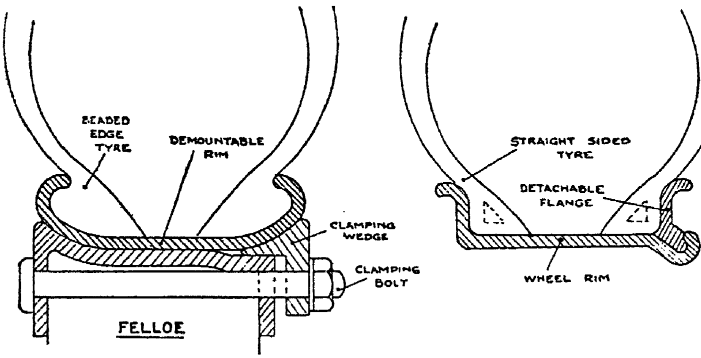


FIG. 287.—(Left). Detachable Rim Type of Wheel, showing bolt and wedge.

(Right). The Straight-sided Tyre Rim, with Detachable Flange.

Fig. 288 illustrates the construction of a modern type, namely, the Dunlop one, and shows the principal features. The relative thickness of the tread is indicated by the black section portion. Between the tread and the inner cord casing are two reinforced breaker layers separated from the tread and casing—and from each other—by layers

of rubber for cushioning purposes. The breaker tread affords a sound protection against puncturing.

The *beaded edge* tyre shown on the left in Fig. 287 was

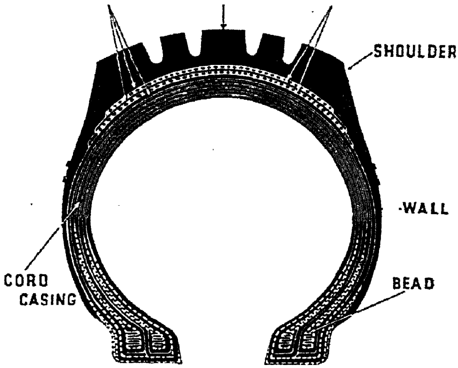


Fig. 288.—Modern Tyre Construction.

at one time far the more popular type. It enabled an inexpensive continuous rim to be employed, but necessitated stretching the tyre with special tools in order to get it over the rims.

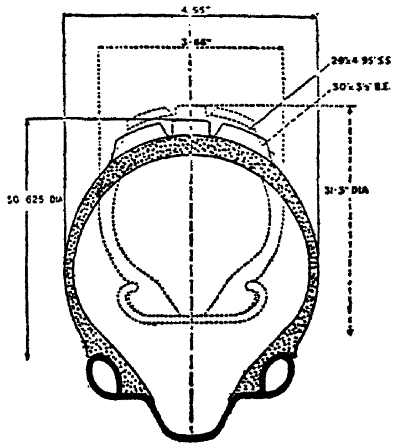


Fig. 289.—Showing the respective proportions of Balloon and ordinary Tyres, and also the Well-Base type of Wheel Rim.

The *straight-sided* tyre shown on the right in Fig. 287 displaced the beaded type. In this case the tyre

can readily be slipped off the rim. It is a development of the earlier wired-on tyre, and actually has a wire running through each rim-edge. A special wheel rim is necessary, the rim having a flat or well-base with two raised flanges. Detachable rim flanges were previously used for the straight-sided tyre; in this case only a screwdriver was required to remove the detachable rim member. The straight sided tyre is much stronger than the beaded type, has a greatly reduced tendency to leave the rims, is easier to fit, requires no security bolts, gives a greater air tube capacity, and does not 'nip' the inner tube; for these reasons it is much used on ordinary and racing cars.

The inflation pressures of high pressure type tyres range from about 40 lbs. per square inch for a 65 mm., to 50 lbs. for an 85 mm., 60 lbs. for a 100 mm., and 70 lbs. for a 120 mm., tyre section; for tyre loads per wheel of 300, 470, 690 and 1,000 lbs. respectively.

The large section *low pressure* or *balloon tyre* now in common use, enables a much larger air capacity to be obtained for a lower inflation pressure. It has a sectional diameter about 50 per cent. greater than for the ordinary tyre of same overall diameter, and an inflation pressure of about one-half the value; thus, a 30 × 5 in. balloon cover, to replace the 30 × 3½ in. ordinary type, would employ an inflation pressure of 20 to 25 lbs. per sq. in., as against 55 to 60 lbs. per sq. in.

Balloon tyres give greater riding comfort due to their shock-absorbing qualities, and give higher speeds of travel over bad roads; their use reduces the chassis mechanism shocks. On the other hand, special wheel rims are necessary, for the best results, the total wheel weight is greater, the steering system requires re-designing in order to avoid undue resistance to steering action, and a loss of maximum road speed (about 10 per cent. as a rule).

For small and medium sizes of car, their use is a distinct advantage, the suspension being much improved. For larger cars the sizes are apt to become somewhat unwieldy.

Tyre Pressures.—The following table shows the recommended air pressures for Morris cars, ranging from the 8 h.p. to the 25 h.p. models, the wheels being fitted with extra low pressure Dunlop tyres of the well-base rim pattern.

Model	*Tyre size*	*Pressure* *Front*	*Rear*
Eight	4·50—17 in.	26 lb.	26 lb.
Ten-Four	5·50—16	23 lb.	26 lb.
Twelve-Four	5·50—16	23 lb.	26 lb.
Fourteen-Six	5·75—16	26 lb.	26 lb.
Eighteen-Six	6·50—16	23 lb.	26 lb.
Twenty-five-Six	7·00—16	19 lb.	

Later Improvements.—In more recent cars the well-base rim has superseded other types and the low-pressure large section tyre has become universal for motor cars.

The tyre can readily be removed from a well-base rim after deflating the inner tube. It is only necessary

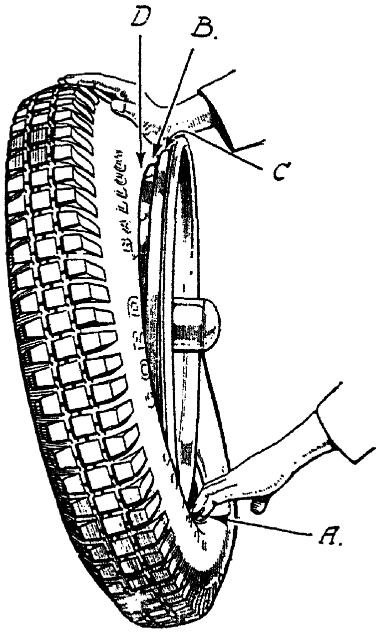


FIG. 290.—Removing a Low Pressure Tyre from Well-base Wheel Rim.

to remove all valve parts and then to push both of the outer cover edges into the well of the rim at the part diametrically opposite the valve; the cover edges are then levered over the rim edge about 3 or 4 inches from the

valve with the aid of two small tyre levers (Fig. 290). *The tyre is fitted* by pushing one edge of the cover over the edge of the rim, the first part put on being pushed down into the rim well. The tube is then inflated slightly and placed in the cover with the valve through the hole in the rim. The second edge of the cover is next fitted and, commencing at a point diametrically opposite the valve, the edge is pushed down progressively into the base of the rim. Small levers should then be used to ease the last few inches over the rim edge.

Before the tyre edge at *A* (Fig. 290) can be pulled over the wheel rim edge the tyre edges at *B* must be pushed over the rim shoulder *C* down into the well *D*; the tyre edge *A* can then be moved over the rim edge, easily.

Wide Rim Tyres. A more recent tendency in tyre design is to employ the wider base rim pattern for low pressure tyres. It has been shown that an increase in the width of wheel rim for a given size of tyre increases the lateral stability and cornering power of the tyre. It has the same effect as an increase in inflation pressure but it reduces the radial flexibility. In regard to the effect of tread life on rim and inflation pressure the following are some results published by the Goodyear Tyre Co.:—

Calling the tread life with a 65 per cent. rim width and 28 lb. pressure 100 per cent., the life with 70 per cent. rim width and the same pressure is 114 per cent., with 75 per cent. rim width and 27 lb. pressure, 119 per cent., and with 80 per cent. rim width and 26 lb. pressure, 122 per cent.

These figures were averaged from results obtained with three tyre sizes, six different cars of three different makes, on routes with all sorts of topography, and with tyres either kept in the same position on the car or changed occasionally between the four wheels.

Wider rims make the car more stable in operation, or free from 'wander,' and the results indicated that a 1-in. increase in rim width is roughly equivalent to an increase of 1·5 lb. in the inflation pressure in this respect. This may be compared with the effect on stability of an increase in the tyre size, from 6·00 to 6·50 in., which was found to be roughly equivalent to an increase of 6 lb. in the inflation pressure.

The beneficial effects of wider rims are partly offset by

accompanying disadvantages. Widening the rims makes the car harder riding, and an increase in the proportional rim width from 65 to 80 per cent. is roughly equivalent to an increase of 3 lb. in the inflation pressure in this respect. This is the reason why lower inflation pressures are generally used in tires on wide-base rims. Also, with the wider rim the flange may be more exposed to kerb damage.

Tyre Developments. A later development in tyre construction is *the use of rayon,* or artificial silk, cords instead of cotton ones. These cords are only one-third of the weight and about 10 per cent. stronger; moreover there are more cords in every ply of fabric as they are thinner than cotton. As the rayon cords are continuous and smooth there is less internal friction when the tyre

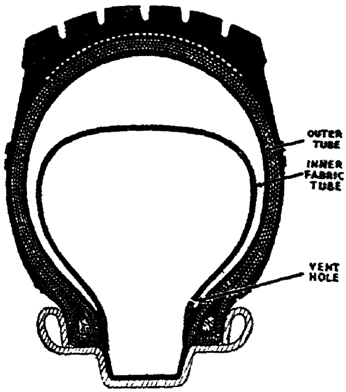


FIG. 291.—The Lifeguard Tube.

flexes so that less heat is developed and the tyres normally run cooler than cord fabric ones. A four-ply rayon tyre has greater strength and fatigue resistance than a six-ply cotton tyre but weighs about 10 per cent. less.

An extra inner tube, known as the Lifeguard, introduced into this country from America is shown in Fig. 291. The object of this is to convert the hitherto dangerous effects of a tyre burst into a harmless slow puncture, thus giving the driver of the vehicle ample time to stop without any risk of skidding or loss of control. It consists of an additional inner tube made of strengthened

fabric, arranged inside the ordinary tube. It is of smaller section than the other tube and floats freely, being placed in communication with the larger tube by means of a small hole in order to equalise the air pressures. Should the outer tube burst the air from the outer chamber only is lost, whilst the inner tube—which holds about two-thirds the amount of the original total air takes the weight of the vehicle on the particular wheel affected. This air escapes slowly through the vent hole, about 20 minutes being taken to deflate the inner tube entirely. Drastic tests made with nails and knives on the tyre have proved its efficiency.

A recent method of constructing tyres introduced by the

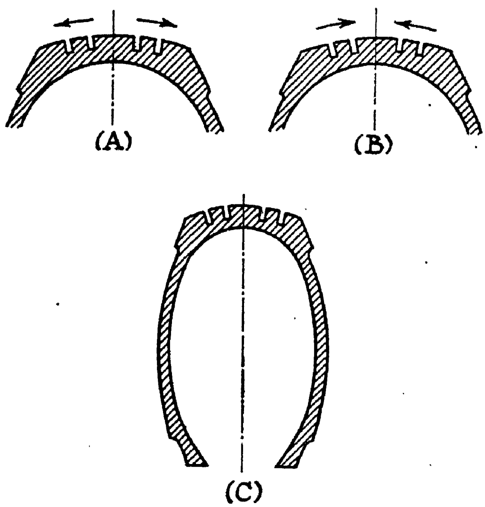


FIG. 292. Compression Tread Design of Tyre.

Goodyear Company is based upon the principle that when rubber is under compression and receives a cut or abrasion the two sides close together, whereas if the rubber is in tension it tends to open out and spread. These effects are shown in Diagrams *B* and *A*, respectively, Fig. 292. The newer method of tyre construction illustrated in Diagram *C* results in the tread of the tyre being under compression when the tyre is inflated. The tyre is of the flat-sided elongated shape when in the non-inflated condition, but when it is inflated the side walls expand and

the tread contracts thus placing the tread under compression. It is claimed that this tyre has a 33 per cent. greater wearing life than the conventional pattern; increases the cushioning quality; adds to the stability of the car and reduces the rolling effect on corners. The diamonds of the tread pattern have been placed closer together in order to increase the non-skidding effect. In addition, riding ribs are provided on the shoulder of the tyre in order to take the greater part of the wear and to eliminate stones and other solid matter that might cause tread abrasions.

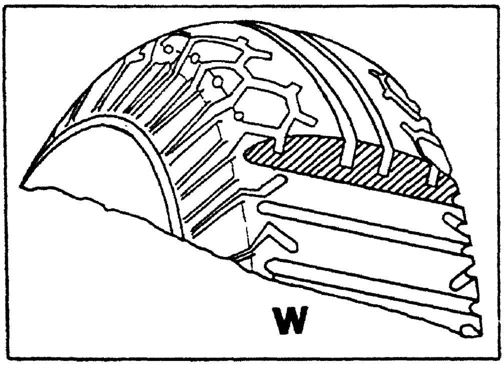


FIG. 293.—The Avon Duo-Tread Tyre, showing at W, the appearance of the Worn Tyre Tread.

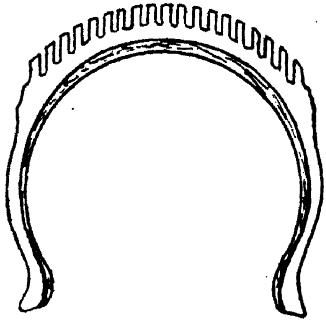
Fig. 293 shows the Avon-Duo tyre which is of special construction, having a series of transverse holes through the tread, such that when the ordinary non-skid pattern tread has worn down the holes then become transverse grooves which present a new non-skid tread on the tyre; this method overcomes the usual disadvantage of the smoothness of ordinary worn tyres.

It is now possible, however, to treat smooth tyres in such a way that peripheral non-skid grooves or ridges are cut in their treads by means of special tyre-grooving machines marketed for the purpose.

There is also a special design of non-skid tyre on the market (Figs. 294(A) and 294(B)) which adopts the same principle as the peripherally grooved tyre obtained by the method previously mentioned. The numerous rubber fins

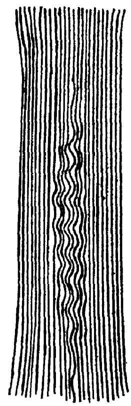
of the tyre offer a high resistance to lateral forces and thus to skidding tendencies.

Tyres intended for high speed track racing cars are usually fitted with smooth treads instead of patterned



(A)

FIG. 294.—Non-Skid Tyre.



(B)

Appearance of Tread under Lateral Action.

ones, in order to reduce the rate of wear that otherwise would occur and to obviate any risk of the tread separating from the casing. The tyres run cooler with smooth tread tyres.

A white rubber strip is often fitted under the smooth tread in order to act as an indicator to the track observers or pit attendants that wear has occurred and the tyres need changing.

Considerable progress has been made in rubber compounding, in recent times, with the result that the wearing and resistance to deterioration under weather conditions have been improved considerably. The wear resistance of rubber has been shown to increase with the amount of carbon black in the rubber. Previously about 20 to 25 per cent. of carbon black was used, but in modern tyres the rubber treads contain from 40 to 50 per cent. of this pigment.

The tendency of tyres to crack on the treads and walls has been reduced by anti-oxidants ; the latter also improve

the flexing qualities of the rubber and prevent premature ageing under sunlight conditions.

A good deal of progress has also been made by employing different cord angles for the various classes of tyres, since tyres for different classes of service require definite cord angles. If the cord angle is increased the tyre becomes more rigid but is liable to injury to a greater extent than for smaller cord angles. The higher the cord angle the shorter the length. Thus, if the cord angle is lowered the flexibility and cord length both increase.

Puncture Proof Inner Tubes. From time to time in the history of tyre development attempts have been made to render the inner tubes puncture proof. One effective method that has been used in recent times for the tubes of commercial vehicles is to thicken the section of the tube which comes under the outer cover tread in order to increase its resistance to penetration by loose road objects such as long flints or nails. The puncture seal type of tube now in use is of the compression type in which an inner rubber layer vulcanized or cured in position is turned inside out before splicing in order to put compression on the inside cured member of the tube.

Another method for puncture sealing tubes is to introduce a special compound into the interior of the tube so that in the event of a puncture this will effectively seal the hole. The compound is usually of a thick syrupy nature and is forced into the hole by the tube air pressure.

Bullet Proof Tyres. The puncture-proof principle has been developed and applied to the tyres of British military motor vehicles so that bullets entering or passing through the tyres do not cause any appreciable loss of pressure, the tyres being self-sealing in this respect.

Liquid-Filled Tyres. In the case of heavy vehicles used for cross country purposes such as tractors, the tyres are often partly filled with a liquid mixture in order to increase their adhesion to the ground. A typical mixture consists of 15 per cent. calcium chloride and 85 per cent. water. About three quarters of the tyre is filled with this liquid, which does not freeze until its temperature is about 20° Fah. below zero. This method is stated to give

better track adhesion, smoother rotation of the wheels and improved driving steadiness of the vehicle. The liquid can be filled and also discharged through the tyre valve.

Causes of Tyre Wear.—The principal causes of motor tyre wear are as follows:—(1) Under-inflation. (2) Excessive road speeds. (3) Violent acceleration and braking. (4) Misalignment of the wheels. (5) Presence of oil or grease on the tyres.

The most usual cause of excessive wear is under-inflation which results in causing the sides of the tyre, or its walls, to bend sharply as the wheel revolves until the cords eventually break, and the tyre blows out.

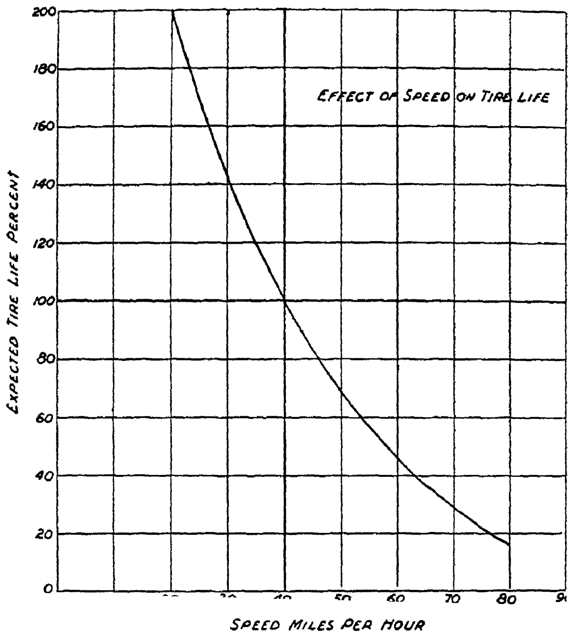


FIG. 295.

In regard to the effect of road speed upon tyre wear, the results shown in Fig. 295, show how the tyre life diminishes as the car speed increases. Thus, at 80 m.p.h., the tyre life is only about one-fifth of that at 40 m.p.h.; at 60 m.p.h. it is about one-half of that at 40 m.p.h.

Misalignment of the front wheels, resulting in the wheels turning in or out too far, is also a frequent cause of excessive wear. The wheels are given a small amount of "toe-in" as explained, previously, and this value should be checked at intervals, more particularly if the front of the car, or the wheels have received a heavy impact. The correct alignment can be obtained by adjusting the length of the track rod, by the screwed adjustment provided for this purpose.

The periodical reversing or interchanging of the front tyres is recommended as a means of extending their lives, for the outer edges of the tyre usually wear more quickly than the inner ones.

Tyre Valves.—The valve fitted to the inner tube is of the non-return type opening inwards against the action of a small compression helical spring. Fig. 296 shows a part-sectional view of the type of valve fitted to many earlier and existing tyres. It has a red-rubber conical seal *B* held in place by a small screw having two projections. The valve cap *A* has a reinforced rubber washer which beds down against the top of the valve body thus making an air-tight joint. The inner non-return valve at the upper end of the compression spring,

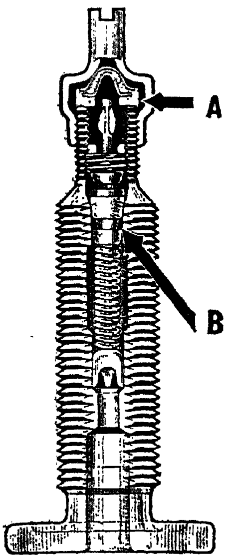


FIG. 296.

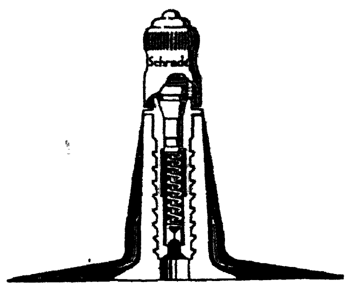


FIG. 297.

shown in Fig. 296 actually prevents air escape from the inner tube, the air-tight cap being an additional safeguard. Incidentally, the cap has a slotted stem for engaging with

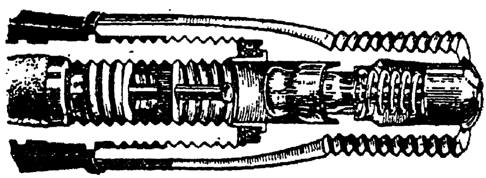
the two projections on the screw holding *B* in place so that the latter with the non-return valve and spring can be removed bodily, for tyre deflation purposes.

In the case of large wheel hub, low pressure tyres a somewhat different type of tyre valve (Fig. 297) is used. This has a rubber base and stem in order to provide flexibility for the tyre inflator connection and at the same time to give a neat and compact valve unit. The stem is generally placed obliquely to the plane of the wheel; it has a similar valve and cap to that previously described.

The valve shown in Fig. 296 is attached securely to the inner tube by means of a steel plate washer and a nut, the enlarged lower end of the valve being inside the tube. The valve unit shown in Fig. 297 is attached to the tube by the vulcanising process, the metal stem having a number of serrations for gripping the rubber shank.

A drawback to the ordinary types of tyre valve is that it is necessary to remove the valve cap before one can test the tyre pressure or inflate the tyre; and to replace the cap afterwards.

FIG. 298.
The Dublchek Tyre Valve.



In order to overcome this disadvantage, a new design of tyre valve cap, known as the Dublchek (Fig.29 8), has been placed on the market by Messrs. Schraders, Birmingham. The valve cap in question has a second valve which provides a further airseal and thus protects the valve core against air leakage and dirt or water entry.

The valve cap in question replaces the ordinary one and with this type there is nothing to take off when testing or inflating a tyre, the pump connector being attached directly to the valve cap.

Bearings.—One of the chief reasons why the engine and transmission of the modern car are so efficient, is that the frictional losses of power in the various bearings have been

reduced to minimum proportions. Plain bearings are used where the loads are heavy, and there is insufficient room for the larger diameter ball or roller bearings.

It is usual to employ plain bearings for certain chassis members which merely rock, when working; for example, the spring shackle pins, steering swivel pins, steering gear, brake and clutch mechanism pin joints. Usually the pins are hardened, and in the best practice work in gun-metal bushes; grease or oil nipples are fitted, the pins being drilled and grooved for this purpose.

In order to reduce the amount of power lost by friction in the continuously rotating components of the transmission, ball bearings are employed for the medium to light loads and roller bearings for the heavier loads. In this connection the co-efficients of friction of properly mounted and lubricated ball and roller bearings are of the order of 0.002 to 0.003, whereas for well-lubricated gunmetal or phosphor-bronze bearings the value is 0.01 to 0.02, i.e, about 5 to 10 times as much as for the former types of bearings; it will be evident, therefore, that a good deal of the power loss with plain bearings can be avoided by substituting ball and roller ones.

Ball journal bearings, except those of the deep-groove type, will not take much end load, so that ball thrust bearings must be fitted where side forces act on journal bearings. Roller bearings take much heavier loads than ball bearings, since the rollers have line, instead of point contact. The two types most commonly used are the Timken tapered roller bearing and the Hyatt helical roller bearing; an example of the use of the latter was in the case of the earlier Ford back axle. By employing two tapered roller bearings, with their axes inclined to each other, end loads from either direction may be taken. The Timken bearing is now widely used for front and rear axles.

The Hyatt bearing has a number of flexible rollers wound from chrome-vanadium steel rod, running in a hardened and ground bearing, or shell; the flexibility of the rollers enables them to adapt themselves to inequalities.

Detachable Wheel Attention.—It is important when replacing detachable wheels, or rims, to see that the surfaces which come in contact are perfectly clean, and

that all the security bolts and nuts are screwed up evenly and tightly. Each nut should be screwed up by hand as far as possible straight away; then a turn should be given to each one, progressively and consecutively, with a wheel nut spanner until all are tight.

After the car has been running for a time, the wheel nuts should be tested, as they are apt to slacken a little. More particularly is this the case with detachable wheel rims. Slackness of the rims frequently causes an irritable creaking noise, resembling that of rusty springs. The threads of the bolts should be greased, preferably with a graphite compound; the tyre rims may also with advantage be rubbed with graphite.

Axle Lubrication.—The spaces in the hubs of the front axle wheels, between and around the ball or roller bearings are usually filled with soft grease, the axle hub cap being screwed home so as to form a dust-right joint. After long periods of running, the grease-retaining felt washers on the inner ends of the stub axles become worn, and allow the lubricant to escape; the former should be replaced when this occurs. The front axle hubs should be refilled or replenished with a suitable automobile axle grease, about every 2,500 to 3,000 miles of running. All nuts on the back axle casing joints should be checked for tightness; slack nuts will cause a loss of lubricant. Advantage should be taken of the opportunity, when replenishing the front hub lubricant to inspect and adjust the bearings.

CHAPTER X

TRACTION AND BRAKING PRINCIPLES

It is proposed, to consider, briefly the principles connected with the traction and braking of automobiles, for the benefit of the technically inclined reader.

Power at Road Wheels.—If the horse power developed in the engine cylinders be represented by 100, then, owing to the power absorbed, or lost due to friction of the piston, the bearings and engine gears, the h.p. available at the flywheel, or clutch, will only be about 87. There is a further power loss in the gearbox due to gear wheel friction and the churning action on the lubricant; this is a maximum on low, and minimum on top, or direct gear. Further losses of power occur in the universal joints, final drive, differential, and between the tyres and the ground. It may be accepted that wherever two gears are in mesh, there is a power loss of from 4 to 5 per cent. Taking everything into account, the h.p. available at the road wheels for propulsion purposes, will be about 70 to 75 on top gear, and about 60 to 65 on low gear. If the engine brake h.p., *i.e.*, at the flywheel, be taken as the reckoning point, these figures will be 80 to 86, and 69 to 75 respectively; these values are actually percentages, and represent the transmission efficiencies.

Power for Propulsion.—The horse power available at the road wheels is used up, during propulsion of the car, partly in overcoming the road friction (rolling resistance) and partly in overcoming wind resistance. Under level road conditions, the following formula shows the power required to overcome these resistances:—

$$\text{h.p. to propel car} = a + \frac{WRV}{375} + KAV^2$$

where W = weight of the car in tons, R* the road

* For pneumatic tyres on smooth macadam, $R=30$ to 35; for solid rubber tyres, 50 to 60. For soft wet roads, $R=60$ to 100 for pneumatic tyres.

resistance in lbs. per ton, V the car's speed in m.p.h., A the cross-sectional area of the car, a and K being constants. The second term represents the power absorbed in road resistance and the third that lost in windage.

The formula can be written as follows :—

$R=a+bV+cV^2$ (lbs. per ton weight of car) where a, b and c are constants depending upon the design of car, size and shape of its body, type of tyres and road resistance.

The power required to propel a car along a level road is given as follows :—

$$\text{H.P.}=\frac{R}{550}\cdot\frac{22}{15}\cdot V.W$$

where $\frac{22}{15}\cdot V$=car speed in feet per second and W=weight car in tons substituting the value for R, previously given we have :—

$$\text{H.P.}=\frac{1 \cdot 46}{550}V\ (a+bV+cV^2).$$

It is possible from the engine performance curve to predict the acceleration and hill-climbing abilities of the car and its maximum speeds on each of the gear ratios provided by the gear-box. A full account of the method employed is given in the Author's book on *Testing of High Speed Internal Combustion Engines* (2nd Edition) (Chapman and Hall, Ltd., London).

Power Required to Climb Hills.—When a car is driven up a gradient more power is required, as the weight of the car has to be lifted through a vertical distance. If the gradient be expressed as 1 in 4, say, this means that for every 4 feet the car moves it rises vertically by 1 foot. The additional power required to climb a hill of 1 in 4 at V m.p.h., with a total weight of W tons is given by

$$\textit{Extra}\text{ h.p. to climb gradient of 1 in 4}=\frac{2240WV}{375\times 4}$$

$$\begin{array}{l}\text{In general, the extra h.p.}\\ \text{for a gradient of 1 in } n\end{array}=\frac{2240WV}{375n}$$

The *total* h.p. required to propel the car, includes that required to overcome the road resistance, windage and gradient, so that the last term given above must be added

to the other two terms in the preceding paragraph in order to obtain an expression for the total h.p. of propulsion.

Adhesion and Braking.—The total weight of the car is taken by the four wheels; there is usually more load on the rear wheels.

If the centre of gravity of the car is at G (Fig. 299), then the weight on *each* front wheel will be $\frac{W}{2}\cdot\frac{b}{a+b}$

and on *each* rear wheel $\frac{W}{2}\cdot\frac{\quad}{\quad+b}$

When the car is on a downward incline, there is less weight on the rear wheels than when it is on the level road. The same formulæ will apply, however, if we

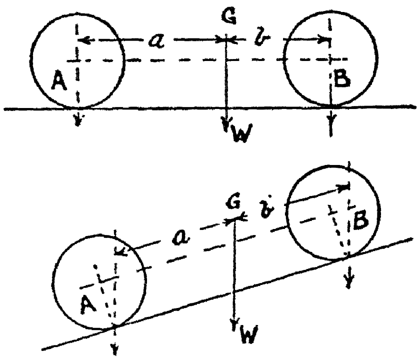


FIG. 299.

reckon the distances a and b *between the vertical lines shown*. (Fig. 299).

When the rear road wheels are on the ground, any attempt on the part of the transmission to turn them will result in the wheels travelling along the ground. There is, however, always present a tendency on the part of the tyres to slip without travelling forward. If there is little load on the wheels, and a great driving effort due to the transmission, they will skid or slip without moving along. The force required, at the tyre, to cause this slipping, is termed the *Adhesive Force* (sometimes the Adhesive Power, or Adhesion). It depends upon the

coefficient of friction, μ, between the tyre and the road, and the weight on the wheel axle. If the latter weight be denoted by the term W, then we have

$$\text{Adhesive Force } P = \mu \cdot W.$$

For the ordinary grooved or studded rubber tyres on dry macadam roads, the value of μ may be taken as 0·6, so that $P = 0{\cdot}6W$.

The value of W may be obtained from the expression given earlier in this paragraph.

Thus, for each rear wheel we have

$$P = \frac{0{\cdot}6W}{2} \cdot \frac{a}{a+b} = \frac{\qquad}{a+b}$$

Now, the ***greatest force which the brakes of a car can exert on any wheel*** is governed by the adhesive force; when the braking force just exceeds the latter, the wheel will slide along, or 'lock.'

It is thus evident why front wheel brakes add to the braking power of a car, by utilising the adhesion of the front wheels.

To ***increase*** the braking force on the rear wheels, it is necessary to load the wheels more (by throwing the C.G. backwards), and to provide non-skid tyres (for which μ has a greater value). When the car is ascending a gradient, the load on the rear wheels is increased, and, therefore, the adhesion also.

Car Deceleration and Braking Relations. The following formulæ represent the relationships between the stopping distances, deceleration rates, braking efficiencies and road resistances. They are based upon measurements of the times or distances of stopping a car from a given road speed. V=speed of car in m.p.h. S=stopping distance in feet from V m.p.h. t=stopping time from V m.p.h. in seconds. f=rate of retardation, i.e., the deceleration in feet per second, per second. d=deceleration in m.p.h. per second. P=road resistance to car's motion in pounds per ton weight of car. A=angle of inclination of road, in degrees. E=braking efficiency, per cent.

Then $$E = \frac{V^2}{0{\cdot}3S} = \frac{4{\cdot}55V}{t} = 3{\cdot}1f = 4{\cdot}55d = \frac{P}{22{\cdot}4}$$
$$= 100 \text{ sine } A.$$

The latter relation refers to the braking efficiency necessary to hold a car on a gradient of angle $A°$. The other formulæ refer to level road conditions.

Also. Deceleration $f = \frac{V}{t} \times \frac{22}{15}$ ft. per :

Stopping distance $S = \frac{Vt}{2} \times \frac{22}{15}$

$$= \frac{1}{2f}\Big(V \times$$

Torque at Rear Wheels.—The mean turning moment of the engine at the crankshaft is given by the relation :—

$$\text{Torque} = 5252 \frac{h.p.}{N} \text{ lbs. ft.}$$

where h.p. is the brake h.p. at the crankshaft and N the revs. per minute.

Assuming a gear ratio of 5 to 1 at the rear wheels, the torque at the rear wheels (assuming a transmission efficiency of 75 per cent.) will be :—

$$\text{Torque} = 0{\cdot}75 \times 5 \times 5252 \frac{h.p.}{N} \text{ lbs. ft.}$$

In the general case, for a gear ratio (gearbox and final drive) of n to 1, and a transmission efficiency of e per cent. we have :—

$$\text{Torque at (two) Rear Wheels} = 5252 \frac{e \cdot n \cdot h.p.}{N} \text{ lbs. ft.}$$

Tractive Effort. The tractive effort at the rear wheel tyres and road can be estimated from the engine torque if the gearbox and back axle gear reduction ratios are known, as follows :—

$$\text{Tractive effort} = \frac{T \times G.R. \times A.R. \times e \times 12}{r} \text{ lbs.}$$

where T = mean engine torque, in lbs. ft.
e = overall transmission efficiency
r = radius of tyre (rolling value) in inches
$G.R.$ = gearbox gear ratio used
$A.R.$ back axle ratio.

Assuming constant values for $A.R.$, e and r this expression can be written as follows :—

$$\text{Tractive effort} = k \times T \times G.R.$$

where k is a constant.

If the engine torque is regarded as constant at its maximum value it will be seen that the tractive effort varies directly as the gearbox ratio. Thus, the greatest tractive effort is obtained on the bottom gear (or reverse gear, if of lower ratio).

Engine Revolutions and Car Speed.—If V is the forward speed of the car in any gear (n to 1), in m.p.h., and D is the diameter of the road wheels in millimetres (1 foot = 304·8 mm.), then it can easily be shown that :—

$$\text{Engine r.p.m., N} = \frac{8538\ V \cdot n}{D}$$

Example.—If a car is travelling at 40 m.p.h. on top speed ($n = 4·75$), and the wheel diameter $D = 800$ mm. we have :—

$$\text{Engine r.p.m.} = \frac{\times\ 40}{800} = 2025.$$

Car Performance Curves. A convenient method of studying the performance of a car is that based upon the engine torque curve, that is, the values of the engine torque plotted against engine speeds over the whole speed range. Values of the torque can be estimated from the relation given on page 353 ; the horse power values are measured at various engine speeds by means of a dynamometer as described in Volume I of this series. It is not required to find the values of the torque on the rear wheels of the car. This can be done by using the formula given on page 353 for each of the gear ratios in turn.

Next, it is a simple matter to find the car speeds corresponding to the engine revolutions given on the engine torque-speed curve, by using the formula given in the preceding section, so that values of rear wheel torque in any gear ratio can then be plotted against corresponding car speeds for all of the gear ratios.

Finally, instead of using torque values, these can be converted into tractive efforts by means of the relationship

given in the previous section on Tractive Effort. In this connection it is more convenient to divide the tractive effort by the total loaded car weight in order to obtain a comparison value, i.e., tractive effort in lbs. per ton.

Fig. 300 illustrates an example of tractive effort performance curves for the four gear ratios given on the diagram in the case of a six cylinder engine car of 18 H.P. rating and 30 cwt. loaded weight.

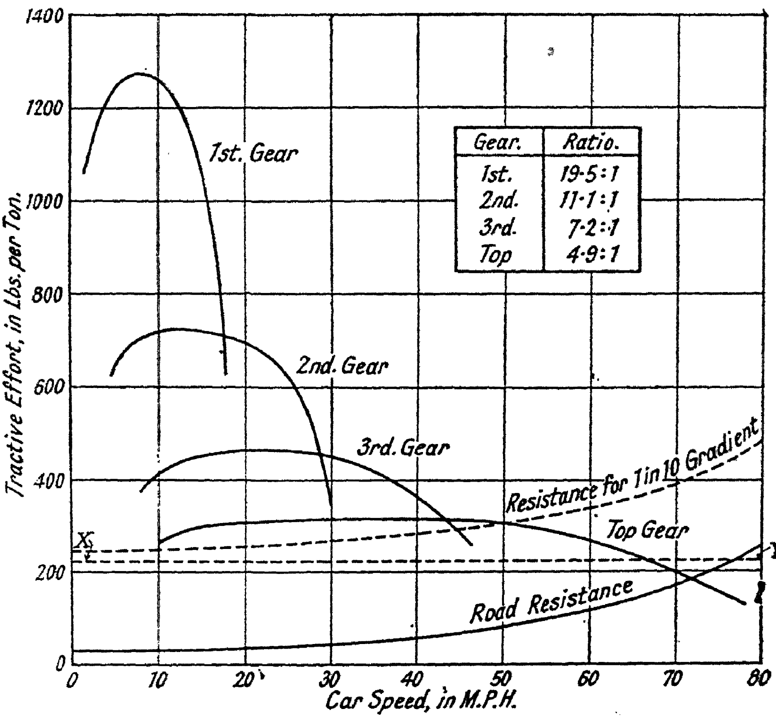


FIG. 300.—Tractive Effort Curves for Different Gear Ratios.

A further curve, showing the average dry smooth road resistance at various speeds is also given; this is based upon the results of road tests.

Each of the gear ratio torque curves represents the available rear wheel torques over the whole engine speed range, so that the extreme right hand portions of the 1st, 2nd and 3rd gear curves indicate the maximum road speeds that can be attained on these respective gears.

Thus, for 1st, 2nd and 3rd gears, the maximum road speeds will be about 17·5, 30 and 46 m.p.h., respectively, for level roads.

Assuming that the car is running on 1st gear at, say, 10 m.p.h. Since the road resistance at this speed is about 40 lbs. per ton, whilst the tractive effort is about 1,230 lbs. per ton, the difference, namely, 1,190 lbs. per ton will be available, either for acceleration from rest or for hill climbing.

If the car were in 2nd gear and running at 10 m.p.h., the net value of the tractive effort for acceleration or hill climbing purposes would be about 670 lbs.; for 3rd gear it would be about 190 lbs.

It will thus be seen that for acceleration at speeds up to 10 to 15 m.p.h. the 1st gear gives the greatest net tractive effort. On this account, also, it will climb steep hills much better.

In regard to the subject of *hill climbing*, the curves given in Fig. 300 enable the best gear ratio to be ascertained for any given gradient. Thus, if a gradient of 1 in 10 be considered, this is equivalent to a tractive resistance of $\frac{1}{10}$ ton per ton, i.e., 224 lbs. per ton. This resistance remains constant, irrespective of road speed, and it is indicated on the graphs by the dotted line *XY* at 224 lbs. per ton parallel to the base line. As the total resistance is the sum of the road and gradient resistances the ordinates of the two graphs must be added together; the upper dotted graph is the result and gives the total resistance for a gradient of 1 in 10.

It will be seen that this graph cuts the top gear curve at 50 m.p.h., so that the car would just be able to climb a gradient of 1 in 10 at a maximum speed of 50 m.p.h. For speeds between 30 and 40 m.p.h., there would be a greater surplus of tractive effort for the 3rd gear then the top one, so that a better acceleration would be obtained. Similarly, for speeds between about 17 and 29 m.p.h. the 2nd gear would give the best performance.

The maximum top gear speed on a level road, as given by the junction of the road resistance and top gear graphs, is 71·5 m.p.h.

CHAPTER XI

FRONT DRIVE CARS

The majority of modern motor-cars follow the same general layout of the transmission members as those produced some twenty to thirty years ago, the arrangement of the front engine and rear-wheel drive remaining practically unaltered. Although this system has undoubtedly benefited by its long period of development, its known drawbacks have spurred a few car designers to attempt to overcome these disadvantages.

The most favoured method is that of placing the engine at the front and driving the front wheels directly, thus leaving the rear wheels to trail behind, as it were.

Among the firms which have produced standard front-wheel drive models, mention may be made of the B.S.A., Derby, Citroen and Alvis in this country; the Auburn Automobile Co., in America; the Tracta, in France; and the Stoewer, Adler and several other cars in Germany.

The original B.S.A. front-wheel drive was a three-wheeled vehicle, the later models are four-wheeled ones.

Advantages of Front-Wheel Drive.—Exponents of the front-wheel drive method make a strong point of the fact that, whereas in the case of the present standard rear-wheel drive the car may be likened to a wagon or carriage that is actually pushed from the back, with the front-wheel driven vehicle the front wheels are positively driven and the rear wheels simply trail behind, as it were. Let us see how this affects the control of the car. In the first case, when a rear-wheel-driven car has to negotiate a corner, the front wheels turn whilst the rear ones are pushed in a direction tending to throw the rear of the car in the opposite direction to the turn, i.e., to cause it to skid. If the car were allowed to swing round freely, it would ultimately take up a position in which the driven, or rear wheels, were actually in front, and the front wheels behind, this being the stable condition of equilibrium. In the case

of the front-driven car, both the power and the steering are applied at the points of contact with the front wheels, the rears wheels merely trailing behind and exerting no appreciable effect upon the control. From both points of view the front-wheel-driven car is undoubedly better to steer round corners or on the straight; moreover, it can negotiate corners safely at a much greater speed than a

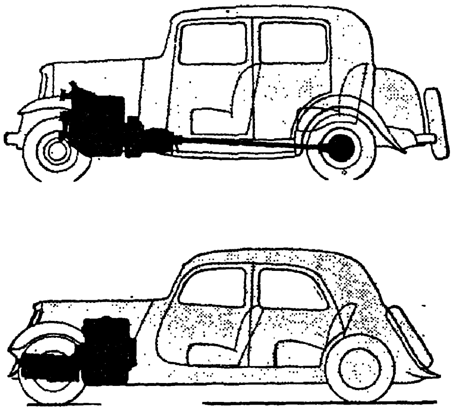


Fig. 301.—Showing (top) Ordinary Rear Drive Car and (bottom) Front Drive Car.

rear-driven car. Racing cars that have been designed with front-wheel drive have distinguished themselves, more particularly in road races where cornering has been an important factor. Another outstanding advantage of front-wheel drive is that it enables a lower centre of gravity to be obtained for the whole car, there being no transmission members or back-axle casing at the rear: Fig. 301 shows, in outline the front and rear drive types of car, the engine and transmission parts being indicated in black. The compactness and low position of the front drive arrangement, with its absence of any driving mechanism between the engine and rear wheels are clearly illustrated. The lower seating position and flat under-shield of the front drive car are other points in its favour; moreover, the body can be made of smaller overall height whilst still giving the same headroom, than in the case of the orthodox rear-driven car body.

It will also be evident that, owing to the absence of any

transmission units between the engine and rear axle and, on account of the lower body that can be used, the front wheel drive car can be made lighter in weight.

The usual arrangement for the front wheel drive is to place the gearbox in front of the engine and to incorporate the final drive bevel pinion and crown wheel drive, as well as the differential gear, inside the gearbox casing. From the differential gear the drive is taken to each road wheel through a short propeller or cardan shaft having a sliding coupling at the gearbox end and an ordinary coupling at the other or wheel end. The universal couplings then operate at wheel speed instead of at 4 to 5 times wheel speed as with the ordinary rear wheel drive propeller shaft. It is a further advantage of this system that the full range of steering angle can be obtained without any difficulty ; in some cases a greater steering lock is possible than with normal rear-driven car steering gears.

Disadvantages.—The chief criticisms which have been levelled at the front-driven cars are that they are of more complicated design and are inclined to lose their driving-wheel adhesion on steep hills. It is true that the front drive necessitates very careful design in order to arrange for the driving, steering and springing of each front wheel. The use of two separate cardan shafts with four universal joints in all, and duplex springs for each wheel is certainly a more complicated one as compared with the ordinary arrangement, but the designers appear to have produced an excellent solution in recent cars of this type. It is a fact however, that, since the drive is on the front wheels, there is a loss of adhesion on hills, due to the more rearward position of the centre of gravity. On very steep hills cases of front wheel spin have actually occurred, due to this factor.

The Adler Car.—This car, of German origin, is made in the litre and 1·7 litre models, each having a four-cylinder type engine.

The arrangement of the front drive unit is along orthodox lines, the drive being taken from the engine through a single-plate clutch to a four-speed synchromesh gearbox and thence to the differential unit which is incorporated in the gearbox front end. From here the drive to the

front wheels is by cardan shafts, each having one fixed and one sliding universal coupling.

The parallel arrangement of transverse leaf springs, with central fixings—similar to that employed upon the Alvis

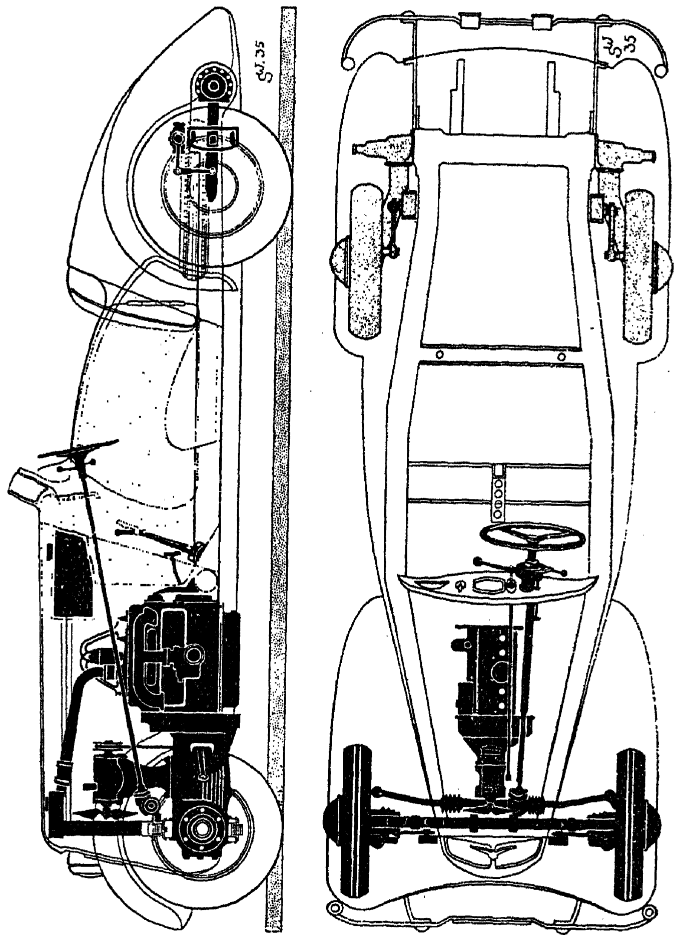


FIG. 302.—Adler Front Drive Car with Rear Torsion Bar Springing.

front drive car—is used, in conjunction with shock absorbers.

The rear wheels are sprung on the torsion bar method, hydraulic shock absorbers being employed; rubber buffers or stops limit the vertical travel of the rear axle member.

Another interesting feature of this car is the arrangement in which both front wheels are — positively from a steering gearbox situated a little

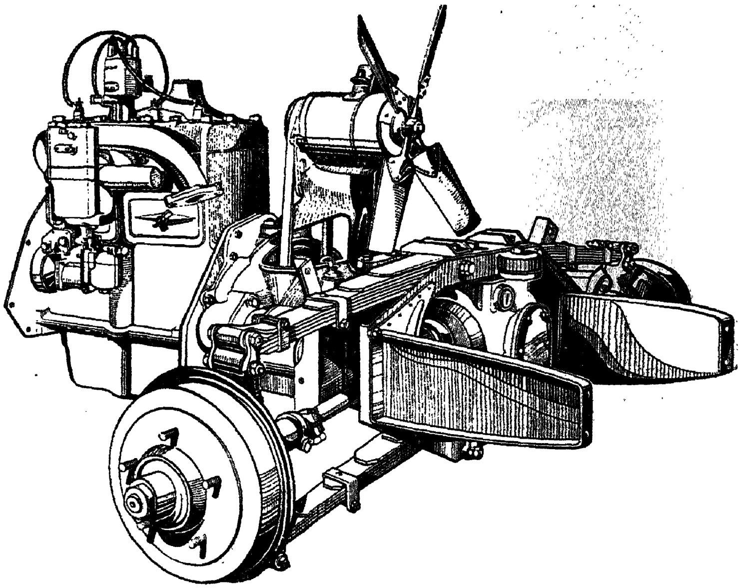


FIG. 303.—Adler Front Power Unit, Drive and Springing System.

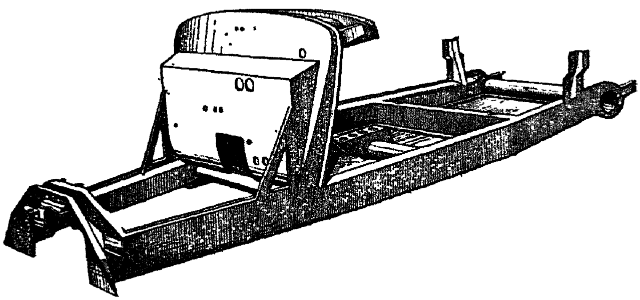


FIG. 304.—The Adler Junior Aluminium Alloy Chassis Frame, with Dash Combined.

centre (Fig. 302). The gear change lever is mounted on the dash board near to the steering wheel, where it is convenient to operate.

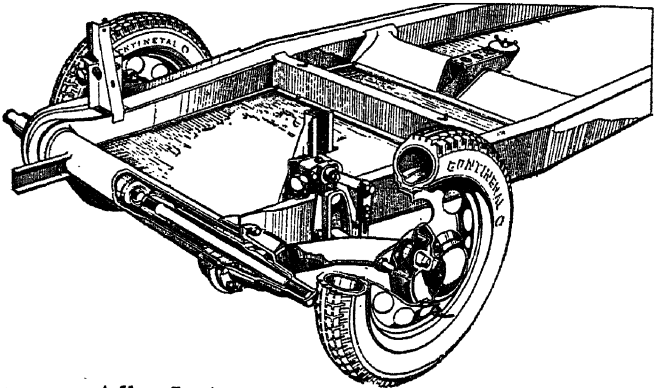


FIG. 305.—Adler Junior Rear Torsion Bar Springing System.

The B.S.A. Front-Drive Car.—This was originally a three-wheeled car having the two front wheels driven from the engine, the single rear wheel being used for steering purposes. Later models have been made with four-cylinder water-cooled engines and two rear wheels, steering then being effected on the front wheels.

The three-wheeled car has a 9 h.p. air-cooled Vee-twin

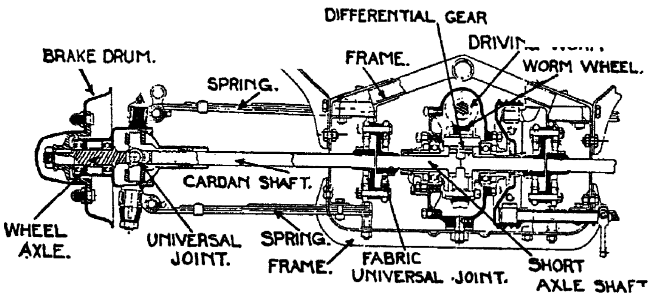


FIG. 306.—Front Sectional View of B.S.A. Front-Drive Car.

engine placed between the driving seat position and the front wheels. This engine drives through a double-plate clutch, running in oil, to a three-speed gearbox arranged in front of the engine. The layshaft of the gearbox is placed above the mainshaft. The latter is extended forwards into the final drive casing and carries a worm, of the overhead pattern, driving a worm-wheel containing the straight pinions of the differential gear-unit (Fig. 307).

The drive is transmitted from the latter to two short shafts, one on each side; these are provided with fabric universal joints connected to the cardan shafts. At the outer or wheel ends of the latter are metal universal joints, taking the drive to the wheels.

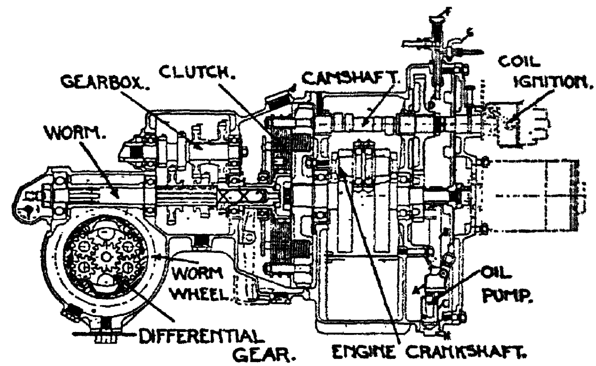


FIG. 307.—Side Sectional View of the 9 h.p. B.S.A. Front Drive, showing Crankcase, Clutch, Gearbox and Differential.

The hubs of the latter are each positioned by four quarter-elliptic leaf springs, one at each corner of the rectangular axle hub units.

The brake-drums are arranged on the wheels themselves.

As the steering in this model is on the single rear wheel, the design of the front-wheel drive and braking system is thereby simplified.

The Citroen Car.—This has been one of the most widely employed of any make of front drive car. It is made in this country in a 12 h.p. model (four-cylinder engine of 1,628 c.c. capacity) and 15 h.p. (1,911 c.c. capacity).

The design of this car is original throughout since it has been a pioneer in regard to the all-steel combined body and frame; detachable front wheel and power-unit section; torsion bar springing throughout and detachable cylinder liners.

The engine, clutch, gearbox and differential all form a single unit which is flexibly mounted to the front section on the 'floating power' principle.

A three-speed gearbox is employed with synchromesh on top and second gears. The differential is arranged

between the gearbox and clutch, the drive being taken through the gearbox primary and secondary shafts and then

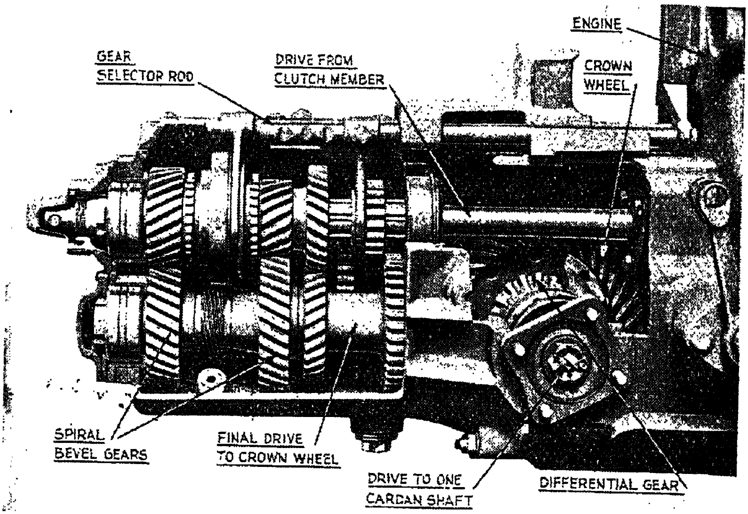


FIG. 308.—Citroen Gearbox and one Final Drive Unit.

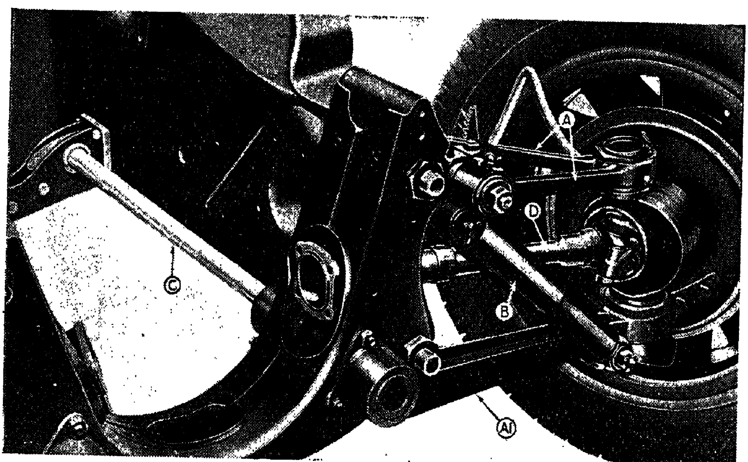


FIG. 309.—Citroen Springing System and Cardan Drive Shaft.

back to the bevel pinion; this ensures rigidity of the final spiral bevel drive (Fig. 309).

Referring to Fig. 309, the front wheel independent springing radius arms are shown at *A* and *A*1, the former member being of bifurcated design. The lower arm *A*1 is rigidly secured to the front end of the torsion springing bar *C* which has a bearing in the front chassis frame member and is rigidly held at its rear end. A shock absorber *B* is mounted between the outer end of the lower arm *A*1 and the frame.

The cardan shaft for transmitting the drive to the left hand wheel is shown at *D*; the outer metal universal joint is also seen clearly in the illustration.

Power is transmitted to the front wheels from the differential unit by means of these universally jointed cardan shafts; each of these is also provided with a splined sliding coupling to allow for the endwise movement due to the springing action. The torsional springing method employed in this car has already been described.*

The Cord Front-Drive Car.—This production model car, made by the Auburn Company, of America, has a unit power plant for the engine, transmission, differential, and front brakes in one assembly. An advantage of this arrangement is that there is no long drive shaft and, therefore, tendency to vibrate at high speeds. The engine fitted is an eight-cylinder vertical type (known as the 'straight eight'). This drives through to the gearbox, which is situated between the engine and the radiator, an arrangement which necessitates an original method of changing gears. The gear-lever consists of a long, horizontal, sliding rod, having a ball-end in front of the dashboard; the rod actually slides in a hole through the dashboard. Gear-changing is made in the usual way by sliding the rod and turning it either to right or left to engage the different gear ratios. This arrangement has the advantage of leaving an unobstructed foot space below the dash, as compared with the ordinary gear-lever position. From the gearbox the drive is taken through a suitable reduction gear, embodying the differential gear, to the usual pair of inclined cardan shafts, one on each side.

The springing system comprises a pair of quarter-elliptic springs, separated by a distance-piece, each free end being

* See pages 56 and 58.

connected by means of rubber shackles or brackets on the front axle; it has been found possible to incorporate hydraulic shock-absorbers between the springs to control both up and down action. In regard to braking, the designers have overcome what would obviously be a difficulty, namely, of the action of the brake-shoes in drums attached to the wheels, by placing the brake-drums

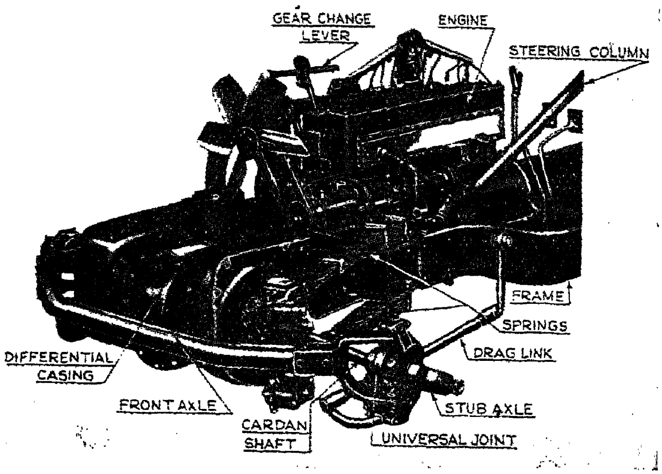


FIG. 310.—Cord Front Drive Arrangement.

on the gearbox side of the universal joint. Although the braking stresses are transmitted through the universal joints, this arrangement is a very convenient one; incidentally, it reduces the unsprung weight by placing the braking gear on the fixed part of the gearbox; Lockheed hydraulic brakes are employed for this purpose.

The front axle of the Cord car is a bent tubular member arranged in front of the transmission unit.

The Rear Engine Car.—In the past, several attempts have been made to fit the engine, together with its transmission, at the rear of the car. The obvious advantage of this arrangement, if it can satisfactorily be carried out, is that more body space can be obtained by dispensing with the bonnet of the car, and also the usual gearbox and propeller-shaft spaces; moreover, there is less noise from the

engine inside the car. On the other hand, the arrangement of the engine, clutch, gearbox and rear-wheel drive as a single unit at the back of the car involves a certain amount of complication, and in some of the examples which have been tested on the road there has been excessive unsprung weight.

The arrangement of the clutch and gearbox controls to be operated from the front of the car is also more complicated than in current practice. Another drawback is that of cooling the engine satisfactorily, for it is difficult to take advantage of the forward speed of the car when the radiator is at the back, so that special fans have to be used in most cases.

In some cases, e.g., the earlier G.W.K. and Trojan cars, the engines were placed in front of the back axle, and drove the latter through friction gearing and roller chains, respectively.

In the case of the later model Trojan car, a two-stroke two-cylinder engine was placed in a compartment at the back of the car—corresponding to the luggage compartment—and drives the back axle.

The Burney 'streamlined' car was another example of a rear-engined car, designed to reduce the noise and heat of the engine. Moreover, the streamlining of the totally-enclosed body enabled the wind resistance of the car to be reduced appreciably, so that higher top-gear speeds were obtained from a given engine output.

The seven-seater Burney car was designed for a top speed of 80 m.p.h., and had an unladen weight of 38 cwt., which figure actually represented an appreciable reduction in total weight for the horse power of the engine due to the design The car was fitted with straight-eight Beverley Barnes engine, rated at 22 h.p., but with a maximum output of 80 h.p. at 3,600 r.p.m. It had a single-plate clutch, the drive being taken through a hollow worm to the gearbox; the worm casing was interposed between the clutch and the gearbox thus lessening the engine overhang beyond the rear axle. A four-speed gearbox was employed ; this actually came in front of the rear axles. There were no continuous front or rear axles, but in front a transverse semi-elliptic springing system was employed. This was used in conjunction with parallel articulated rods fitted with compressed rubber bushes. The springing at the rear was by laminated twin

transverse springs. Each of the wheels was thus sprung independently. The drive was taken by cardan shafts through the gearbox to the rear wheels; radius rods and torque tubes were fitted.

The German Auto Union racing car is a recent example of a rear engine model which has proved successful in road

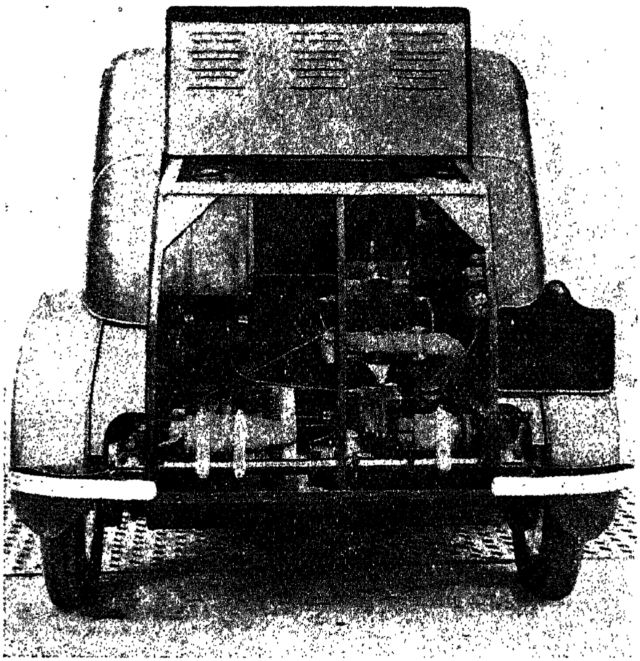


FIG. 311.—The Trojan Rear Engine, Rear-Driven Car.

competitions and on the track. It has a 16-cylinder Vee-type engine of 6-litres capacity transmitting its power to a gearbox arranged behind the rear axle. The torsion bar system of springing is employed at the front and rear; hydraulic brakes are used on all four wheels. The engine is of high output, developing about 600 B.H.P.

CHAPTER XII

LUBRICATION OF THE CHASSIS

Every moving or working member, whether it operates continuously, as in the case of the transmission parts, or intermittently, as with the brake mechanism, should be kept adequately lubricated; further, it should have its bearings protected against the ingress of dirt and water.

Where the parts rotate at normal to high speeds, a lubricating oil should be employed; for example, the engine's bearings, cams and gears, and the gearbox bearings and gears would be oil-lubricated. In the case of slow moving and intermittently moving parts, a heavier lubricant or grease should be used; thus, for the steering mechanism joints, the steering pivot pins, the spring shackles and cardan shaft joints, a medium consistency grease is used. For spring lubrication, a thick grease is best, although some designs of enclosed laminated springs use ordinary lubricating oil; graphite grease is often used for lubricating exposed leaf springs. Finally, the lightly loaded mechanical parts, such as the magneto armature and distributor bearings, the electric generator and and distributor bearings or those of the coil ignition distributor driving shaft, the electric generator and starting motor should be lubricated now and again with a few drops of machine oil.

For the gearbox, as we have stated, lubricating oil similar to that used for the engine is now frequently recommended. With the present-day high engine speeds, it is only logical to include the gearbox with the engine lubrication. Some manufacturers, have hitherto recommended a mixture of equal parts of lubricating oil and gear grease, probably with the object of quietening their gears; a thick lubricant, however, results in appreciable power loss, as modern gearbox efficiency tests have shown; the layshaft should only just be immersed in lubricating oil, for the best results.

The back axle is generally lubricated with a special

lubricant, of a consistency between engine oil and ordinary car grease, supplied for the purpose.

The front axle hubs are packed almost solid with a special axle grease supplied for the purpose ; this should be replenished once every six months of running, as a rule.

An important item to lubricate is the spigot upon which the clutch rotates when it is disengaged ; in many of the earlier designs insufficient attention was given to the lubrication of this part, with the result that much wear and consequent chatter occurred ; with many modern clutches the lubrication of this part is automatically done.

The ball-thrust bearing unit, in the clutch disengagement gear is another important one to lubricate ; this should be oiled frequently, unless the graphite type self-lubricating thrust washer or grease packed bearing are employed.

Chassis Lubrication.—It is one of the principal drawbacks in the public use of motor-cars that there are so many items requiring frequent lubrication when the car is in use ; in some designs one meets, there are no

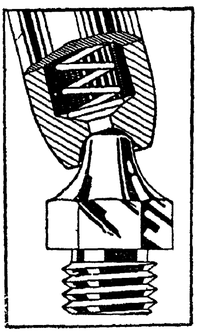


FIG. 312.—Typical Grease Nipple, with Grease Gun Nozzle in Position.

less than twenty different parts to oil or grease regularly. Very often the oil or grease nipples cannot be reached without getting one's hands and clothes dirty in the process.

In order to render the essential lubrication points more accessible in awkward places a short length of metal piping is often led from the lubrication point to a grease gun nipple placed in an accessible position.

In some instances the grouped nipple system is employed.

This method is particularly advantageous where inaccessible bearings are concerned. It consists of a permanent battery plate on which are mounted the greasing nipples, connections from which are made to the various bearings by means of pipe lines; this battery plate is, of course, located in a readily accessible part of the chassis. The method commonly adopted in connection with chassis lubrication is to provide grease nipples of the type shown in Fig. 312. A grease gun containing a relatively large diameter piston—usually from $1\frac{1}{2}$ to 2 ins. diameter—is provided with a small bore outlet and nozzle, such that the lubrication pressure is multiplied up to a high value, namely, from 3,000 to 4,000 lbs. per sq. in. at the nozzle. The latter is designed to fit over the nipple, so that the pressure of the hands on the barrel of the grease gun is sufficient to maintain a good joint between the two parts.

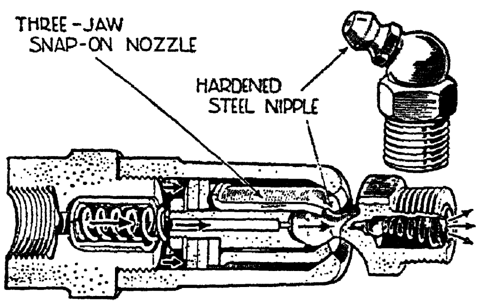


FIG. 313.

An alternative system, due to Tecalemit, is shown in Fig. 313. In this case the hydraulic nozzle fits over an extended pattern nipple screwed into the outer (fixed) bearing and a three-jaw snap-on nozzle holds the nipple securely. The large arrows indicate the path of the lubricant; the smaller arrows show where the pressure forces the chuck forward so as to increase the contact with the nipple.

Another method, known as the Alemite (Fig. 314) employed on many American cars uses a kind of bayonet joint nozzle on the grease- or oil-gun. The twisting action of inserting the nozzle on to the nipple also creates the necessary hydraulic pressure for forcing the lubricant into

the nipple orifice. The nipples have two pin projections similar to those on electric lamp stems.

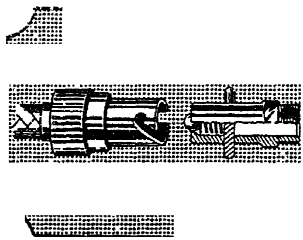


FIG. 314.—Alemite Method of Lubrication.

Automatic Chassis Lubrication.—In view of the large number of 'rocking' type bearings on a modern chassis, the more expensive makes of car are frequently provided

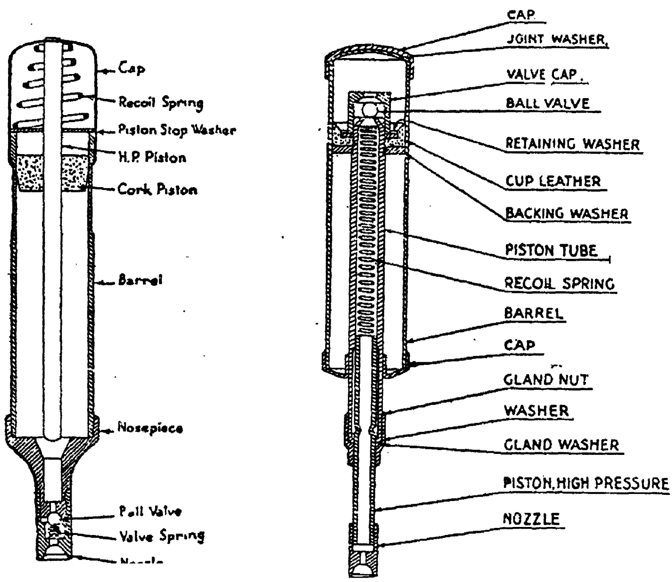


FIG. 315.—Two Alternative Types of Chassis Lubrication Grease Guns (Tecalemit).

with *automatic chassis lubrication* systems whereby oil from a central reservoir under the control of the driver is distributed through a series of fixed and flexible oil-pipes to

each of the bearings of the working members on the chassis. The driver gives a charge of oil to all of these members about once every 100 miles, merely by pressing a pedal or lever, and releasing it.

Fig. 316 shows the Tecalemit pedal-operated central lubrication system.

In this case the central reservoir shown is fitted with an independent pump plunger arrangement for each main outlet from the pump and there is a method of feed restriction at the points to be lubricated. Pumps are supplied

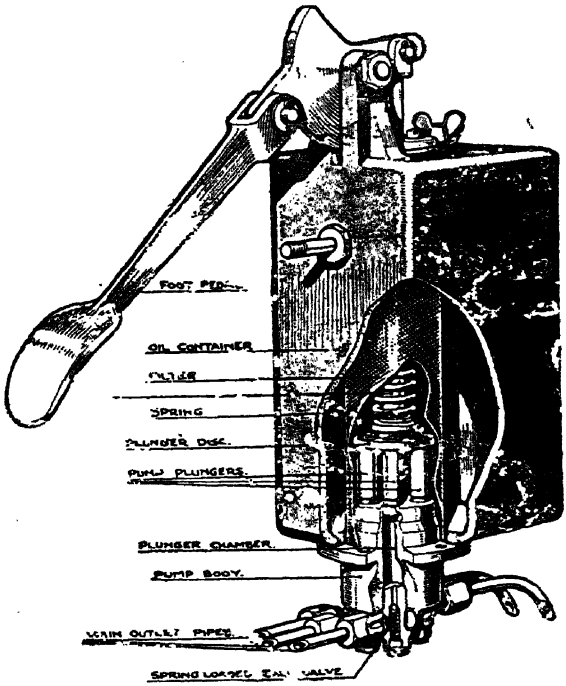


FIG. 316.—Tecalemit Pedal-Operated Chassis Oiling Device.

with as many as five main outlets, each of which discharges lubricant from an isolated chamber at a pressure of 150-200 lbs. per sq. inch to from 8 to 10 points. The lubricant is never wasted because the regulator plugs on the bearings are designed to allow only a predetermined quantity of oil to

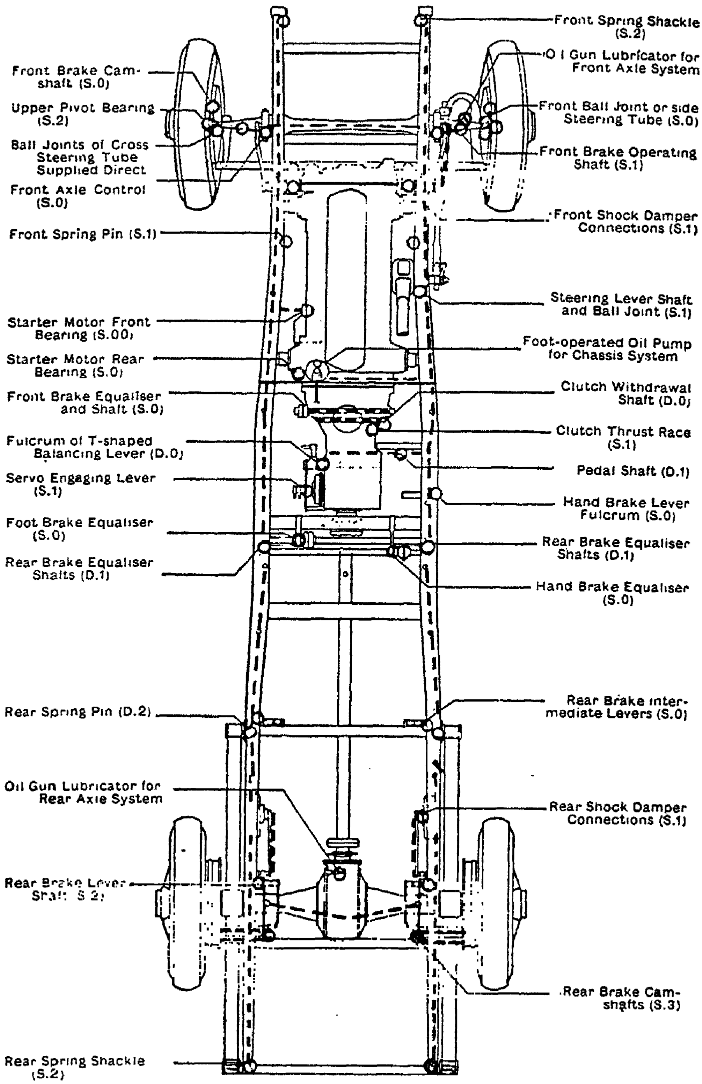


FIG. 317.—The Automatic Lubrication Points of the Rolls-Royce Chassis.

pass in a given time. A large filter inside the container prevents the ingress of any foreign matter to the pump chambers.

When the pressure on the foot-pedal is released, the spring inside the reservoir forces the plungers down and the lubricant is delivered from each chamber through a spring-loaded ball valve to the main outlet pipes. The oil is filtered before entering the pump chambers by passing through a fine-mesh gauze surrounding the main pump-rod spring. The oil thus delivered passes to the various points to be lubricated through special *feed regulator plugs* and *locating bolts*.

The object of the latter is to measure out the quantity of oil supplied to each member lubricated; it is possible to regulate the oil supply to each individual member, in this way.

The *Rolls-Royce chassis* (Fig. 317) is fitted with automatic lubrication on somewhat similar lines, a plunger being provided to give a charge of oil to all the bearings indicated. The capacity of the oil reservoir is such that it equals about 35 strokes of the pump—sufficient for about 2,000 miles. When the engine is started-up each day the oil pump should be depressed twice.

Automatic Pump Device.—There is another chassis lubrication system which is so arranged that the driver is relieved of the necessity of working any plunger; this operation is performed automatically by the car's motion.

In this system, so long as the vehicle is in motion, oil is pumped through the system under high pressure and admitted to the bearings in predetermined quantities. The main unit is a very small pump which discharges one drop of oil for about every 40 or 50 strokes of the pump at a pressure of about 250 lb. per sq. inch. The moving parts of the pump are case-hardened and work submerged in oil.

The pumping unit (Fig. 318) is encased in a glass bowl with a removable metal cap. This bowl holds approximately a quart of oil, which is enough to last about 2,000–3,000 car miles or 1,000 lorry miles. A larger container can be supplied if required; this is made of pressed steel and holds about 3 quarts of oil. The metal cap is easily removed when the container requires refilling.

Under this cap is hinged an inertia weight balanced on a

spring. This weight responds, by oscillation, to the movements of the vehicle, and when in motion, operates a floating plunger-rod, to the lower end of which is connected the pumping unit situated in the bottom of the container.

Oil enters the pump when the inlet port is uncovered by the oscillatory motion caused by the car's movements on the road, and as the inertia weight falls, the plunger forces the oil downwards past two valves. The pressure main-

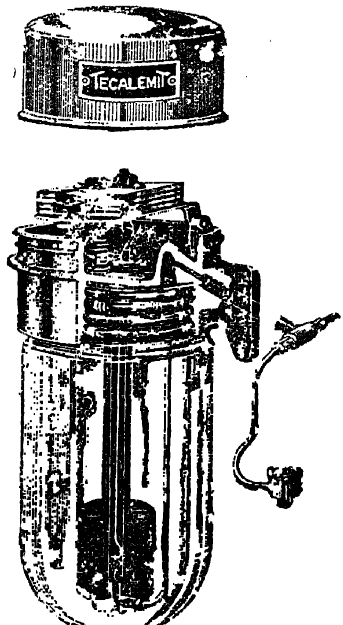


FIG. 318.—Automatic Chassis Oiling Device.

tained by continuous pumping distributes the oil throughout the system, forcing it to the bearings through the feed regulator plugs. The high pressure developed results from the very small diameter of the plunger in the pump.

The Luvax-Bijur System.—Known as the Luvax-Bijur system this automatic method of chassis lubrication utilizes the engine oil pump as the source of pressure, oil being drawn from the engine sump and fed to the chassis bearings, through a special oil feed regulator which controls the amount of oil flowing through. This regulator is

connected to the pressure side of the engine oil pump and it contains a restriction formed by accurately-sized pins fitting with a pre-determined clearance. A filter is included to protect the restriction orifice from dirt. Oil is led from the regulator through pipe lines to the bearings on the chassis. At each of these bearings a meter valve is fitted for the purpose of regulating the amount of oil that is delivered to each bearing.

Fig. 319 shows two alternative types of meter valve of the Tee and straight varieties, respectively.

Each meter valve contains three elements: first, a felt strainer plug S, which strains all oil entering the fitting

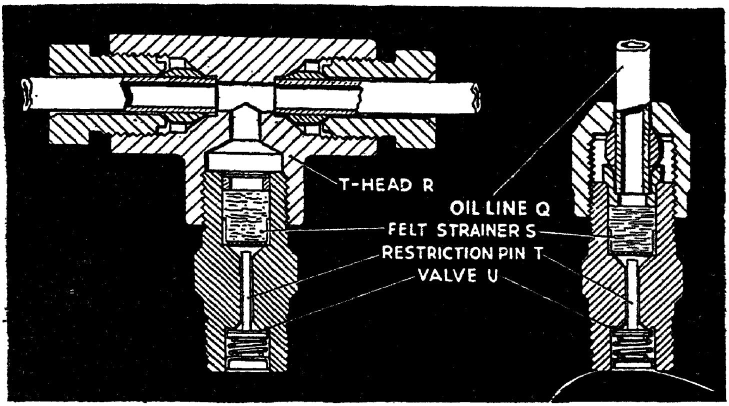


FIG. 319.—Luvax-Bijur System Oil Metering Units.

to protect the restriction crevice and valve from foreign matter by catching such chips, dirt or copper oxide as might originally be in the pipe line. This strainer, Fig. 319, cannot become clogged because all fresh oil entering the piping is filtered by the regulator. The restriction orifice which governs the rate of oil flow consists of a very accurate hole in which lies an accurately sized pin T, so that the clearance between the pin and the hole forms an accurately sized restricted aperture.

In all meter valves the hole is of the same size, but the pins are of different diameters, thus affording different restrictions and consequently different rates of flow. All drip plugs are numbered in accordance with their flow

rate, the numbers ranging from 0 to 6 as this rate increases.

The valve shown at U opens while the line is under pressure from the tank and allows the oil to escape. Whenever the tank pressure ceases the valve closes, and keeps the pipes full of oil ready for the next operation. The meter valve felt strainers catch any loose foreign matter in the pipe lines without being clogged thereby and the felt filter disc in the tank outlet prevents any further dirt from reaching them.

Lubrication Periods for Cars.—Most manufacturers of motor-cars issue instructions in the form of tables, or charts, showing where and when to lubricate the engine, transmission and chassis parts; these instructions should be pasted on to a stiff board, varnished, and hung in a conspicuous place in the garage. The following general notes and instructions, which are based upon a long experience with various makes of car, may be found useful to those who do not possess the tables, charts or other instructions issued by the makers.

Lubrication Instructions.

(1) *Items Requiring Frequent Lubrication (Oil).*
The following parts require lubrication frequently, *i.e.*, every 50 to 100 miles:—The Fan Pulley Bearing. Overhead Valve Rocker Bearings (if not automatically lubricated). The Engine (keep Sump Level correct). The Clutch Withdrawal Bearing and Rollers.

(2) *Items Requiring Frequent Lubrication (Grease).*
The following parts require lubrication, by a grease-gun injection every 100 miles or so:—The Steering Pivot Pins. The Spring Shackles. The Cross Tie Pins. The Spring Trunnions on Rear Axle (when Cantilever or Quarter Elliptic Springs are fitted).

(3) *Items Requiring Periodic Lubrication.*
Oil every 1,000 *miles:* Brake Rocking Shaft bearings (if external). Gear Change Lever Bearing. Clutch Pedal Bearings. Foot Brake Bearings. Hand Brake Bearings. Throttle Control Joints. Accelerator Joints. Ignition Control Joints. Distributor (oil sparingly)

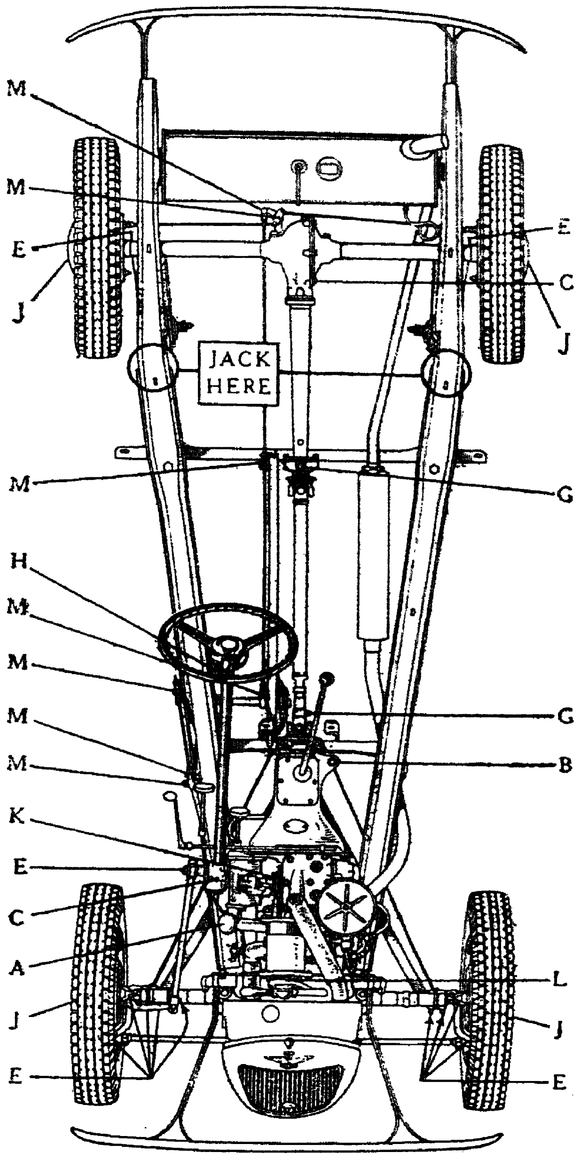


FIG. 320.—Austin Eight Chassis Attention Diagram.

Lubricate every 1,000/2,000 *miles:* Steering Gear Box, Steering Connecting Link and Drop Arm Ball Joints, Universal Joint on Propeller Shaft. Sliding Coupling on Propeller Shaft. Spring Leaves, if of enclosed gaiter type.

(4) *Items Requiring Occasional Attention.*

Oil every 4,000/5,000 *miles*: Back Axle and Differential Casing (with special back axle oil). Gear Box (with engine oil or lubricant recommended by makers). Brake Mechanism joints. Magneto, Dynamo and Starting Motor Bearings. Door hinges and catches.

Grease every 5,000 *miles*: Front Axle Hubs (fill with axle grease).

Every 2,000 to 3,000 miles engine sump should be drained, whilst engine is hot, and fresh engine oil put in.

Every 5,000 miles or so, gearbox should be drained, and fresh lubricant provided. The back axle casing should also be drained and replenished.

Every 5,000 miles or so, graphite or ordinary thick grease should be inserted between the leaf spring blades, if these are not enclosed in leather gaiters; if dry or rusty inject graphite penetrating oil.

See that all oil and grease holes and passages are quite clear, by inserting a suitable wire. Use the lubricating period opportunities to test all nuts for tightness, and see that all split pins are in position.

Lubrication Diagrams.—As previously mentioned, practically every manufacturer of repute issues a diagram showing the various parts of the chassis to lubricate and the intervals at which this operation should be carried out. Some parts require more frequent lubrication than others, so that the intervals between the mileages at which these points are attended to are shorter than in other cases. It is usual to give a plan view diagram of the chassis and to indicate by lines, letters or typescript the lubrication data in question. In some cases the makers give a side elevation diagram as well. The methods of indicating the items on the chassis that require attention will be apparent from the two typical chassis lubrication diagrams reproduced.

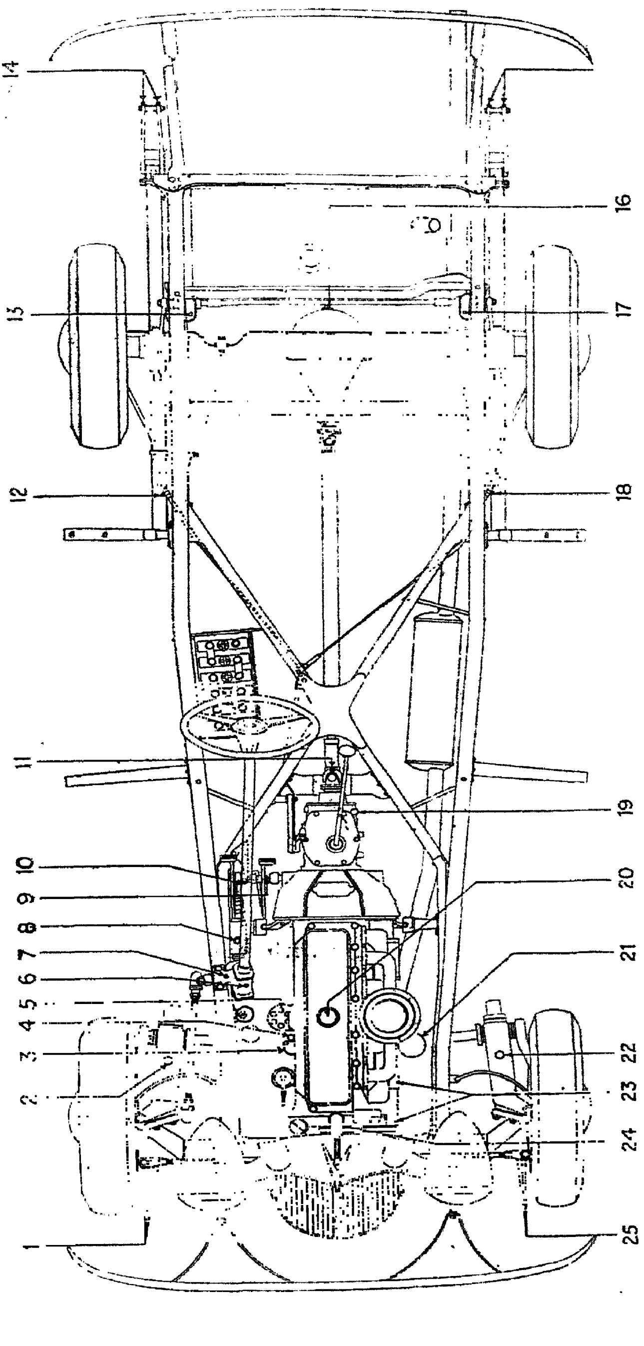


FIG. 321.—The 25 h.p. Vauxhall Lubrication Diagram.

Fig. 320 shows the lubrication diagram of the Austin Eight chassis.

The letters on the chassis plan diagram denote the various lubrication points, the following being the key to these letters :—

A. Crankcase—Replenish to full mark on dip-stick WEEKLY.
B. Gearbox—Replenish MONTHLY.
C. Rear Axle and Steering Box—Replenish MONTHLY—Special Oil.
E. Steering Cross Tube (2) Steering Side Tube (2), Swivel axles (2), Spring bushes (6). Grease WEEKLY.
G. Torque tube, front end, Propeller shaft, splined end. Grease MONTHLY.
H. Top of steering column—Oil MONTHLY.
J. Hubs—Grease MONTHLY.
K. Distributor—Oil sparingly every 1,000 miles.
L. Fan Bearing—Grease MONTHLY.
M. Brake and throttle control joints, Oil WEEKLY.

Fig. 321 shows the lubrication diagram of the 25 h.p. Vauxhall car, the following being the corresponding instructions for the parts indicated by the figure numbers :

DAILY—OR EVERY 200 MILES

(4) Check engine oil level and replenish through filler (20).

WEEKLY—OR EVERY 500 MILES

(1, 7, 9, 10, 14, 15, 25) Apply oil gun (filled with medium-heavy-grade gear oil, or light grease) to nipples.

(24) Using oil-can, fill water pump oil reservoir with engine oil.

(8) Check level of fluid in brake cylinder. (This is not for lubrication purposes, of course, but it is vitally important.) Top up only with Lockheed Hydraulic Orange Brake Fluid.

EVERY 2,000 MILES

Drain engine sump and refill with fresh oil at 20.

(3) Clean crankcase oil filter.

(5) Refill distributor reservoir with engine oil.

(16) Add medium heavy grade gear oil as required.

(6, 19) Add light or medium grade gear oil as required.

(23) One or two drops of engine oil only.

EVERY 4,000 MILES

(2, 13, 17, 22) Add special shock absorber oil as necessary.

(11) Give two or three strokes with oil gun.

(12, 18) Lubricate with Acheson's 'Gredag,' using oil gun.

Remove front wheel hubs and repack bearings with soft cup grease.

EVERY 8,000–10,000 MILES

(16, 19) Drain and refill with fresh gear-oil.

(21) Fit new A.C. filter.

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